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NOTE

In planning this helpful series of Educators, it has been the aim of the author and publishers to present step by step a *logical plan of study in* ***General Engineering Practice,*** taking the middle ground in making the information readily available and showing by text, illustration, question and answer, and calculation, the theories, fundamentals and modern applications, including construction *in an* **interesting** *and* **easily understandable** *form.*

Where the question and answer form is used, the plan has been to give *short, simple* and *direct answers*, limited to one paragraph, thus simplifying the more complex matter.

In order to have adequate space for the presentation of the important matter and not to divert the attention of the reader, descriptions of machines have been excluded from the main text, being printed in smaller type under the illustrations.

Leonardo Da Vinci once said:

> "Those who give themselves to ready and rapid practice before they have learned the theory, resemble sailors who go to sea in a vessel without a rudder"

—in other words, "*a little knowledge is a dangerous thing.*" Accordingly the author has endeavored to give ***as much information as possible*** in the space allotted to each subject.

The author is indebted to the various manufacturers for their co-operation in furnishing cuts and information relating to their products.

These books will speak for themselves and will find their place in the great field of Engineering.

Contents of Guide No. 2

CHAPTER 14

CORLISS ENGINES

Introductory.—When mathematicians were investigating the expansive action of steam, and the saving that might be thus effected, a professor in Providence, R. I., looking over some of the calculations, became interested, and took them to a young man who had shown inventive ability while working at the harness makers trade. The latter was George H. Corliss, who then turned his attention to the steam engine, and invented a valve gear for using steam expansively. The great success of this gear made Corliss famous.

FIG. 348a.—First design of Geo. H. Corliss.

The first engine designed by Corliss with circular valves, the kind in general use now, was constructed in the year 1850. The expansion of steam had been tried with poppet valves and a fixed cut off, but had not met with much success. With the Corliss gear, the cut off can be varied and controlled by a governor; thus the degree of expansion is automatically changed to suit the load conditions.

Conditions Necessary for High Efficiency.—Economy in the use of steam requires that it shall be:

NOTE.—The first engine fitted with the Corliss gear was of the beam type with flat slide valves. There were separate inlet and outlet ports, which were made as short as possible. The valves gave a rapid admission and cut off, the latter being obtained by releasing weights, suspended from a lever.

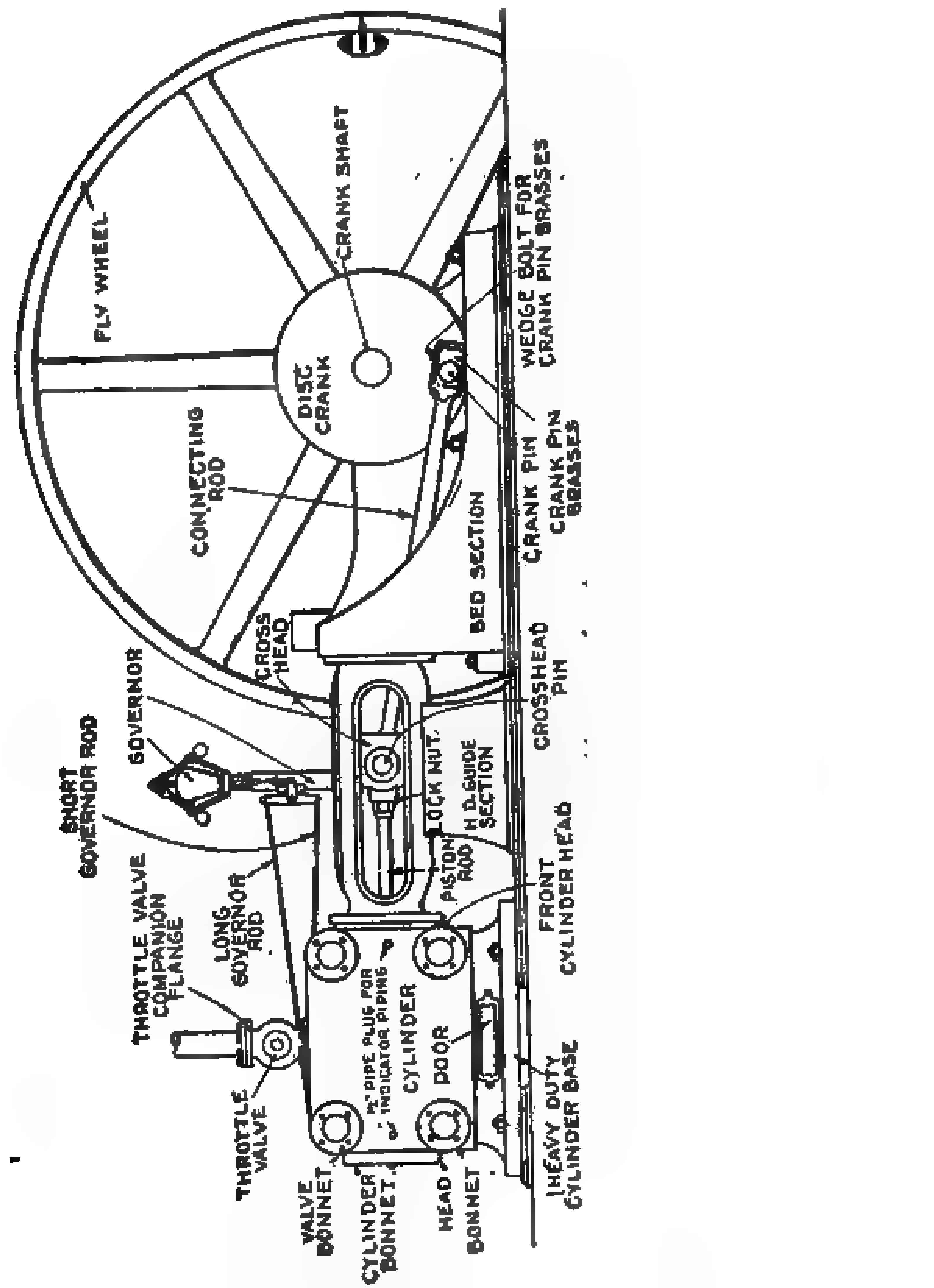
FLY WHEEL
CRANK SHAFT
WEDGE BOLT FOR CRANK PIN BRASSES
DISC CRANK
CRANK PIN
CRANK PIN BRASSES
CONNECTING ROD
BED SECTION
CROSS HEAD
CROSSHEAD PIN
GOVERNOR
SHORT GOVERNOR ROD
LOCK NUT
H D GUIDE SECTION
PISTON ROD
FRONT CYLINDER HEAD
LONG GOVERNOR ROD
THROTTLE VALVE COMPANION FLANGE
½" PIPE PLUG FOR INDICATOR PIPING
CYLINDER
DOOR
HEAVY DUTY CYLINDER BASE
THROTTLE VALVE
VALVE BONNET
CYLINDER BONNET
HEAD BONNET

FIG. 880.—Corliss engine construction; plan with names of parts.

1. Quickly admitted to the cylinder at as near boiler pressure as possible.

2. Cut off with great rapidity quite early in the stroke.

In general, the most economical cut off is from one-seventh to one-fifth stroke condensing, and from one-fifth to one-fourth stroke non-condensing. The cut off should be rapid to reduce *wire drawing*.

FIG. 331.—A Corliss cylinder, with lagging, valves, etc., removed to show construction. The transverse cylindrical chambers at the end are for the valves; those for admission being on top, and those for exhaust at the bottom, thus affording natural drainage. Steam and exhaust passages connect each pair of valves; the exhaust passage being placed at a distance from the cylinder so that the exhaust steam at comparatively low temperature will not absorb heat from the steam within the cylinder and thus cause condensation before release. The long stroke gives small clearance.

3. Pre-released as late as possible without increasing the back pressure;

The full benefit of the available expansion is thus obtained.

FIGS. 852 and 853.—Vilter high pressure, and low pressure cylinder of large Corliss compound engine. *In construction,* the cylinders are of semi-steel with double ports for both steam and exhaust valves. The steam valve is of the T type, the exhaust valve being of the cored box type, and both are actuated by T headed valve stems. The cylinders are jacketed with magnesia or other insulating material.

4. Exhausted with a minimum of back pressure;

5. Compressed only sufficiently to absorb the momentum of the reciprocating parts, and fill the clearance space, at a pressure most suited to the conditions of operation;

6. Used in an engine having very little clearance.

The small clearance of the Corliss engine is one of its economic features, being only 2 or 3 per cent of the cylinder displacement. In some short stroke high speed engines, the clearance is as much as 10 per cent.

M.

FIG. 854.—Bilgram diagram illustrating some defects of the slide valve; *1*, excessive travel when designed for short cut off; *2*, reduced port opening at short cut off with shifting or swinging eccentric; *3*, premature release and compression at short cut off with shifting or swinging eccentric. These defects have been explained at length in the chapter on the slide valve.

Defects of the Slide Valve.—The economic steam distribution described in the preceding paragraphs is not obtained with the slide valve on account of its limitations, previously explained.

The chief defect of the slide valve is its inability to handle steam expansively, except to a small degree. The valve itself cannot cut off shorter than about six-tenths stroke without excessive travel. When operated with a movable eccentric, early cut off is obtained at the expense of reduced port opening. Moreover, early cut off is attended with premature release and compression; at still earlier cut off these defects become more marked. One may be corrected by changing the inside lap, but this involves a correspondingly larger error for the other.

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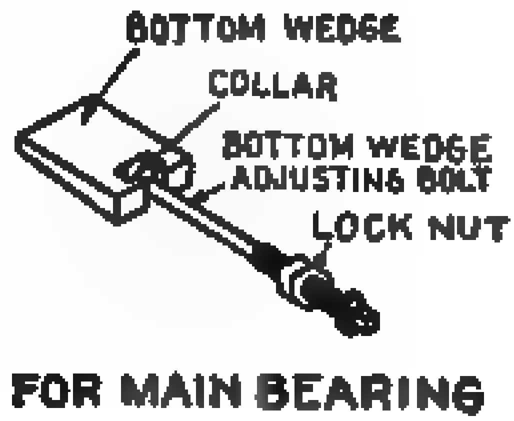
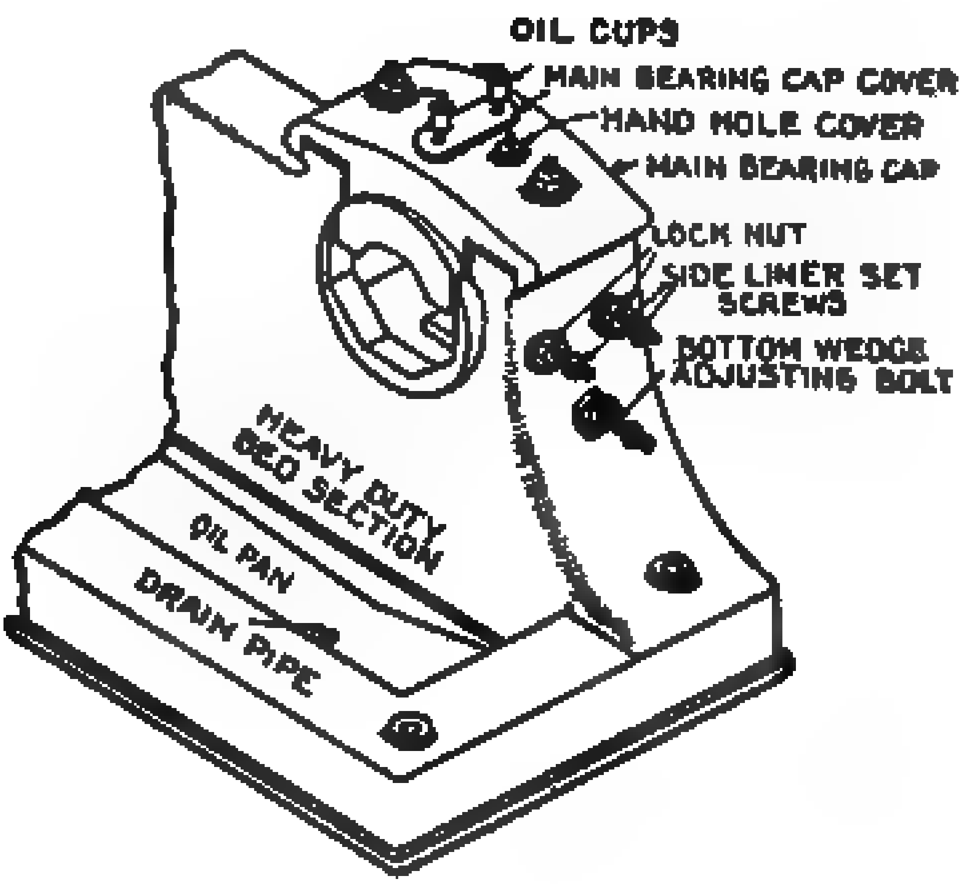
FIG. 855.—Sectional view of a Corliss cylinder showing construction and giving names of parts.

NOTE.—In the *Corliss valve gear*, one important poin ... ving the valve small movements when closed, and large, and therefor ... e power required to drive the valve gear to a minimum. For instance ... through the first and second half of its total arc, the steam admission ... st valve 11° and 27° respectively. For ordinary engines of this type,



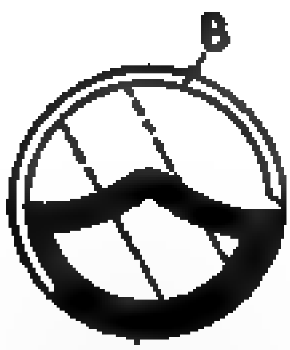


FIGS. 856 to 864.—Details of Corliss engine construction. Figs. 856 to 861 piston details; figs. 862 to 864, end of frame with details of main bearing.

Corliss Valves.—The limitations of the slide valve are practically overcome with the Corliss gear. As shown in fig. 851, there are four valves, two at each end of the cylinder. Steam is admitted by the upper valves, figs. 865 and 866, and exhausted by the lower ones, figs. 867 and 868, this arrangement reduces the clearance to a minimum, and provides proper drainage for the water of condensation.

Each valve has one acting edge for the steam; and is shaped as a sector of a circle; it works within a cylindrical seat over

VALVE STEM COLLAR



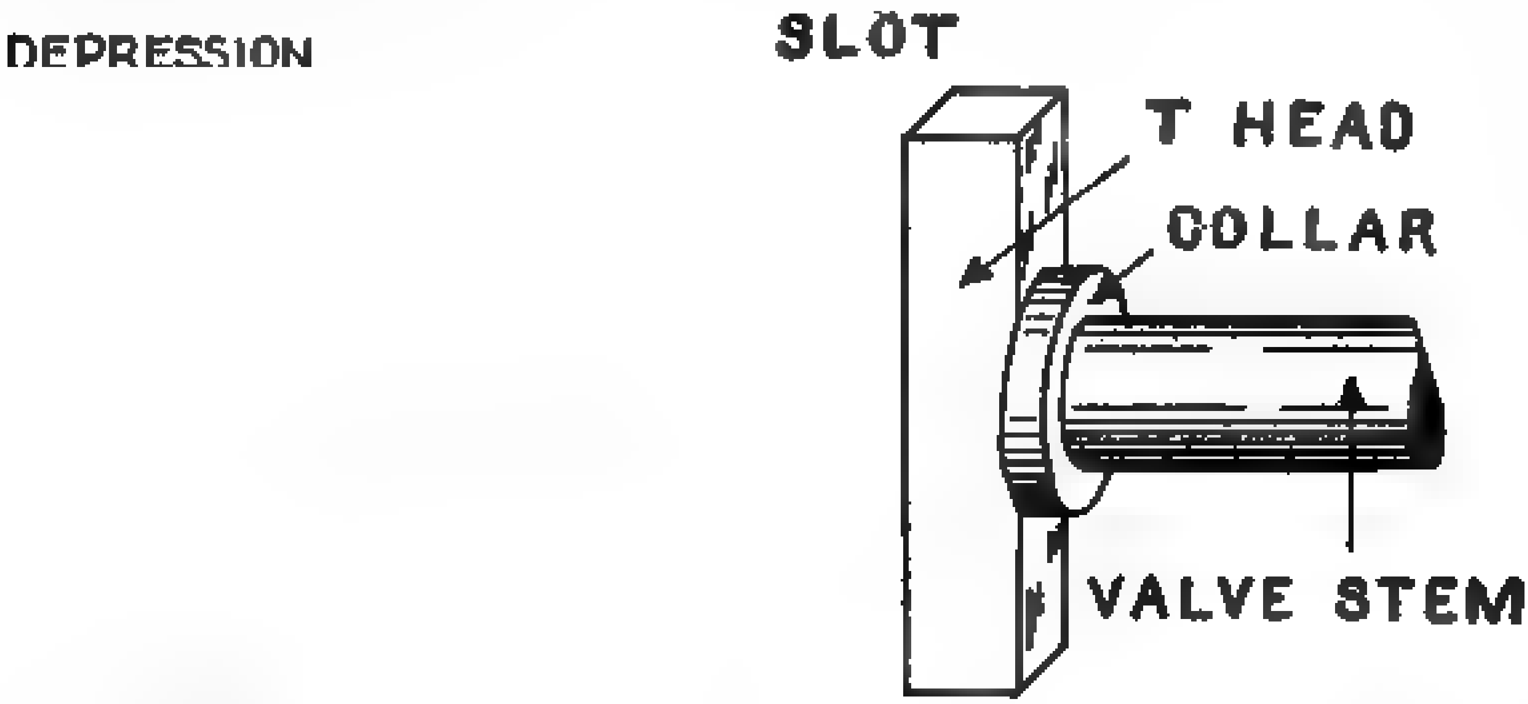
FIGS. 865 to 868.—Corliss valves. They rotate back and forth over circular seats, and are guided by cylindrical ends, whose circumferences are partly depressed, as at A, leaving a space B which allows the valve to raise from its seat and relieve the cylinder of any excessive pressure, as in priming.

ports in line therewith. The valves are made single, double and sometimes triple ported; the usual construction is shown in figs. 865 to 868 which illustrate a steam and exhaust valve.

Each end is cylindrical, fitting the circular end of the valve chamber which acts as a guide. Part of the cylindrical section is cut away to a smaller diameter as at A, leaving a space B, which allows the valve to raise from its seat and relieve the cylinder in case an excess of water be carried in with the steam.

The valve stem has a T head which fits in a slot cut in the valve making a flexible connection. Figs. 869 and 870 show more clearly this construction. A collar forms a part of the valve stem; its object is to protect the bonnet from any cutting action due to the edges of the T head.

Figs. 874 and 875 show two views of the crank end of a Corliss cylinder with valves assembled. The valve stem collar as seen, fits in a circular recess in the bonnet.



FIGS. 869 and 870.—End construction of a Corliss valve and stem. The depression in the cylindrical section of the valve permits it to raise from its seat; there is a slot in the end of the valve for the T end of the stem.

Figs. 871 to 873.—Providence Corliss valves. Fig. 871, double piston inlet steam valve; fig. 872, view of steam valve showing opening edges; fig. 873, triple ported exhaust valve.

SPACE HERF

IN

X

FIGS. 874 and 875.—Two sectional views of a Corliss cylinder showing valves assembled. The small clearance should be noted, also the space for valves to raise from their seats. The space from A, to B, allows access of steam which presses the exhaust valve firmly against the port to prevent leakage.

VERNOR ROD
OOK

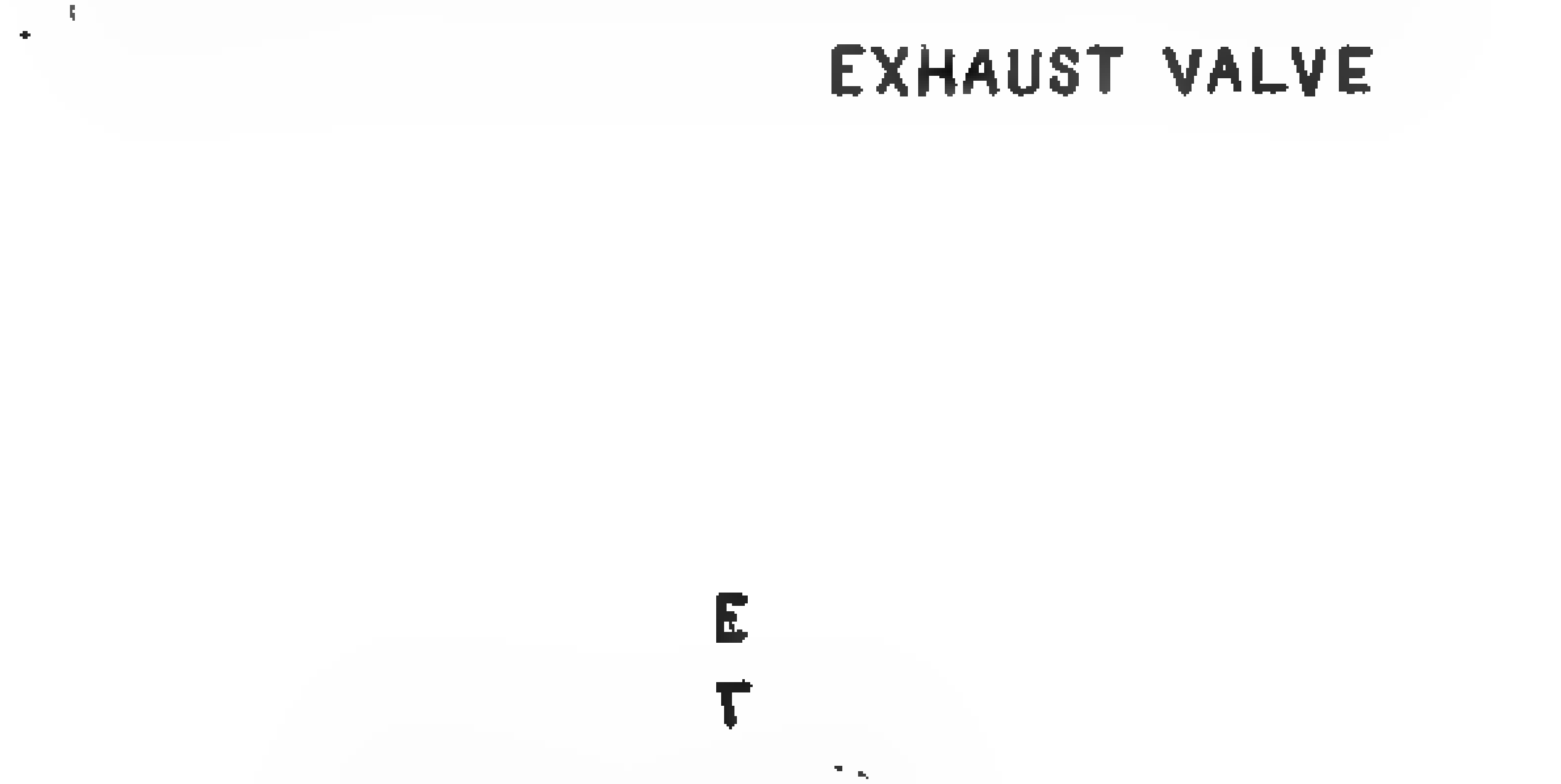
D HANDLE

EXH

FIG. 8
Valv

The exhaust valve is cut away between A and B, giving the steam access to that portion opposite the port, thus pressing the valve firmly against its seat which tends to keep it tight. The valves may be removed from the cylinder by taking off the bonnets C, C′.

In figs. 877 and 878 is shown the end of the valve chambers with bonnets removed; each valve has a threaded hole H, in which a "pull rod" may be screwed to facilitate the removal of the valve. It should be noted that mark P, is scribed on the



FIGS. 877 and 878.—Reference marks for setting Corliss valves. These marks are visible when the bonnets are removed, and show the steam edge of both the valve and the seat.

seat to locate the steam edges of the port, and another V, on the valve to indicate the position of its steam edge. These marks are used in setting the valves, since the ports and edges are not visible.

The Valve Gear.—The mechanism by which Corliss valves are operated is quite different from the plain slide valve gear. For ordinary service, where the maximum cut off is under one-half stroke, all the valves receive their motion from a single eccentric. Heavy duty engines, requiring a long range cut off, have two eccentrics, one for steam valves, and the other for the

YOKE PIN SPINDLE TOP

HEAD

FIG. 880.—The Corliss single eccentric valve gear. Motion ... nsmitted to a wrist plate, and from the latter to the valves ... is given the valves in opening and closing the ports, and a ... e wrist plate is at ... closed periods— ... e names of the various parts are given in the figure.

exhaust valves. Fig. 880 shows the general arrangement of the Corliss gear with a single eccentric. The reciprocating motion obtained from the eccentric is transmitted as usual by the eccentric rod to a rocker, or carrier arm; this arm is connected by a *carrier rod* to a *wrist plate*, or wheel pivoted to an arm which projects from the side of the cylinder.

The carrier rod has a U shaped bend B, which when the gear is connected, engages with a pin P, on the wrist plate. As shown in the figure, the carrier rod is disconnected or *unhooked*. The object of this is to allow the gear to be operated by

hand in starting; after working the condensed steam out of the cylinder, and giving the engine a few turns with the wrist plate operated by hand, the U bend of the carrier rod is allowed to drop on the pin and the engine come to speed; the gear then receives its motion from the eccentric.

When the carrier rod is hooked to the wrist plate, the motion of the eccentric causes the latter to oscillate about its center. Attached to the wrist plate are four rods which transmit motion to the valves. The upper ones are called the *steam rods* because

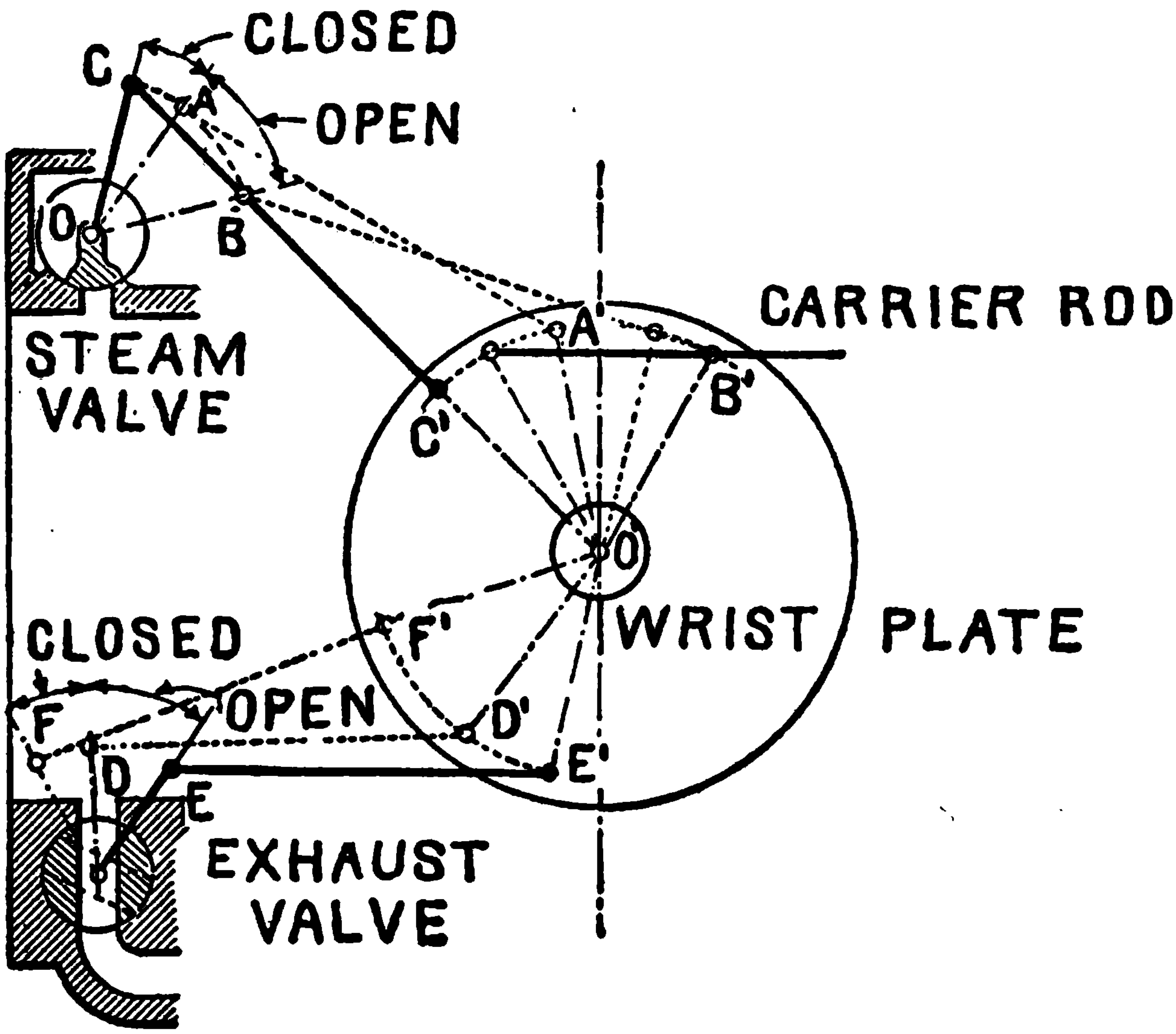


FIG. 881.—Illustrating the peculiar action of the wrist plate and valve rods which produces a rapid opening and closing of the ports, with a slower movement during the closed periods. This reduces wire-drawing, thus utilizing the steam to better advantage.

they operate the steam valves, and similarly, the lower ones are known as the *exhaust rods*.

The success of the Corliss gear is due in part to the peculiar motion given the valve by the action of these rods and the wrist plate. This is illustrated by the diagram, fig. 881, which shows the valves and wrist plate connection at one end of the cylinder.

FIGS. 882 to 894.—Vilter-Corliss valve gear. Fig. 882, steam valve gear; fig. 883, steam valve gear showing knock off bar; fig. 884 to 893, steam valve gear disassembled; fig. 894, dash pot. ***The complete valve gear*** comprises a wrist plate with radial valve rods connecting it to the steam and exhaust valve levers attached to the valve stems, the steam valve stems being fitted with trip blocks, drop levers, knock off levers and drop rods connecting with double vacuum dash pots. The governor cranks are mounted on the valve bonnet, being brought as close to the cylinder as possible. They are provided with fiber knock off cams on top and brass safety cams at the side. The knock off bars are steel forgings, and are keyed to one end of the valve hook stems, on the other end of which are located the valve hook stems, on the other end of which are located the valve hooks or trip pins. The knock off levers are so arranged as to work without springs, but springs are, nevertheless, provided as a safeguard. The trip pins, or valve hooks, as they are sometimes designated, have hardened steel trip plates, with eight wearing edges, which engage corresponding hardened steel trip plates on the dash pot cranks. The trip pins are provided with a special adjustment for regulating the amount of lap of the trip plates. The dash pot cranks are keyed to the valve stems and are provided with pins to which the drop rods are attached. The steam arms are provided with automatic push down or closing pins which engage the dash pot cranks in case of accidental lagging of the dash pot.

In operation, the angularity of the steam and exhaust rods varies in such a manner that the valves move rapidly at the time a quick movement is desirable, as at admission and release, and slowly after the ports have been fully opened, and closed. The latter feature makes it possible to use smaller valves than would be otherwise practicable because, on account of the retarded movement, a given angular motion yields a larger port

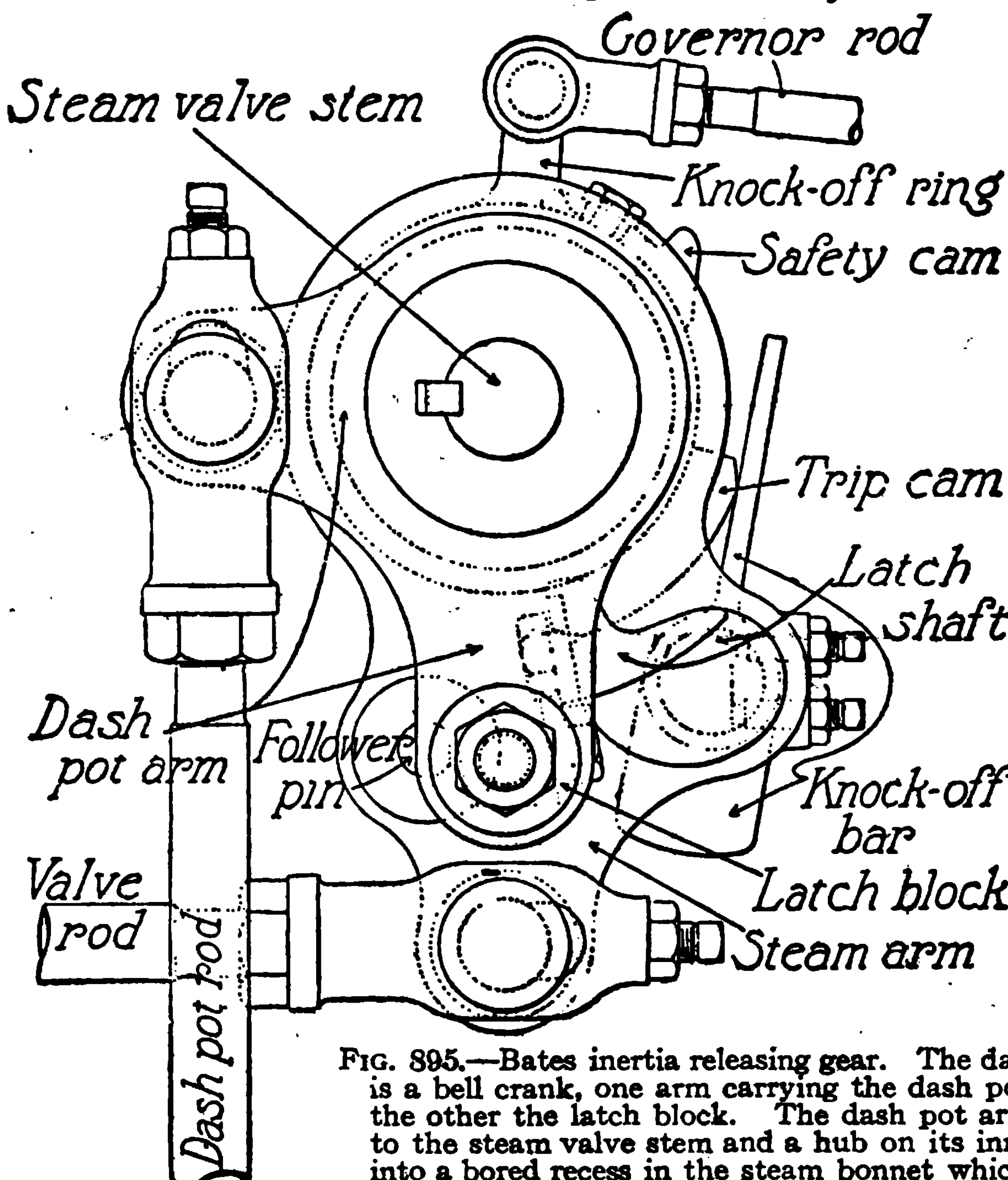


FIG. 895.—Bates inertia releasing gear. The dash pot arm is a bell crank, one arm carrying the dash port rod and the other the latch block. The dash pot arm is keyed to the steam valve stem and a hub on its inner side fits into a bored recess in the steam bonnet which increases the bearing surface. The steam arm is driven by the valve rod in the usual way. A substantial boss carries the latch shaft to which the knock off bar is firmly keyed. The knock off ring is controlled by the governor, through the governor rod, and carries the trip cam and safety cam. The latter only comes into operation in the event of some mishap to the governor, when it is thrown into such position that the valves cannot be opened. In opening the valve, the valve rod moves to the left and the latch shaft engages the latch block in the position shown, and continues in its path until the tail of the knock off bar comes in contact with the trip cam and is wiped outward raising the latch until the dash pot arm is released. The dash pot then comes into action and returns it to the original position. The follower pin is firmly fixed to the steam arm and closes the valve if for any reason the dash pot fail to work. The construction and balancing of the latch shaft and knock off bar are such that the inertia due to the reciprocating motion and the gravity of the parts insures an automatic latching action at the end of the return stroke without the use of any spring, and in turn assists the unlatching at the point of cut off, thereby reducing the reaction on the governor.

opening than could be had with a straight motion of the rod. In the diagram the valves are shown in several positions. It should be noted that as the steam valve opens through the angle A O B, the angularity of the steam rod with respect to the two arcs of motion A B and A′ B′, is at a minimum, hence the motion is rapid, giving a quick admission.

10

BELL CRANK DRIVING ROD

FIG. 896.—The Cooper releasing gear specially designed for high speed. The latching or hooking up operation is performed by a pick up plunger through the action of gravity. The engaging faces of the steel block on the pick up plunger and the steel block on the steam crank are always parallel during their working periods, thus prolonging the wear of the steel block edges.

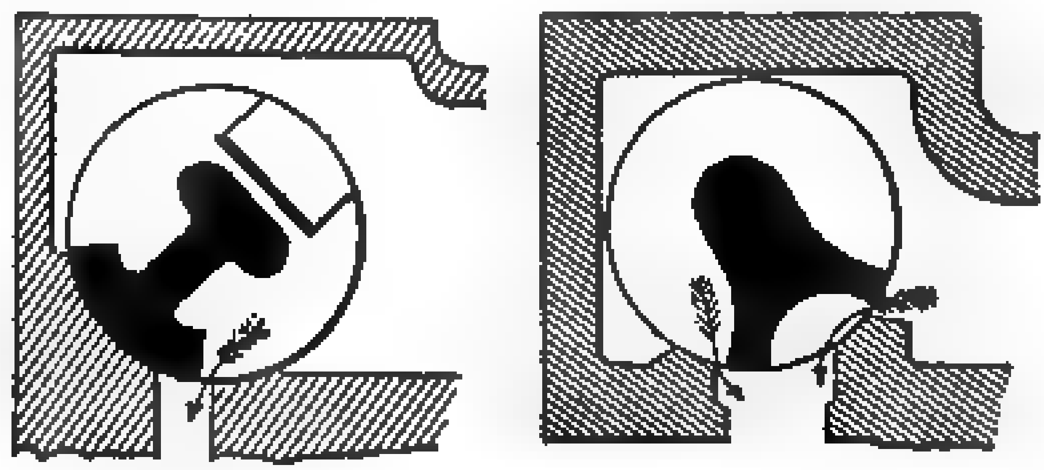
When the wrist plate is moving from C′, to A′, during the closed period of the valve, the angularity is greatest, resulting in a retarded valve movement.

A positive motion is given the exhaust valves, from the wrist plate, but a different method of operation is employed for the

steam valves. The closing motion of the valves at cut off is not positive, but is obtained by a *releasing device* which at the point of cut off disconnects the valve from the main gear and allows it, with the aid of the dash pot, to drop or close with great rapidity.*

This is one of the chief features of the Corliss gear, and its action is both an advantage and disadvantage.

Its good features are the rapid cut off obtained which reduces wire drawing, and thus increases the economy; also the great range of cut off secured without any reduction in port opening.



OUTSIDE ADMISSION — INSIDE ADMISSION — DOUBLE ADMISSION

FIGS. 897 to 899.—Types of Corliss steam valves. Fig. 897 shows an outside admission valve, giving a direct movement with the half moon gear; fig. 898 illustrates inside admission, the valve being operated directly by the oval arm gear. A double admission valve is shown in fig. 899. This valve can be designed to move either way in admitting steam. Outside and inside admission valves may be used with either type of gear, in some cases an indirect eccentric movement is required.

It has the objection, however, that the rotative speed of the engine is limited on account of the non-positive action of the steam valves. High piston speed with this gear, therefore, involves a long stroke, and the slow rotative speed, a large fly wheel. On account of these limitations, a new type or so called "Corliss" has appeared, in which there is direct connections between the wrist plate and the steam valves, the cut off being regulated by varying the travel of the wrist plate motion through the travel of the wrist plate motion through a governor and link.

Engines of this class are properly called *four-valve non-releasing* engines; as they are fully described in a separate chapter.

*NOTE.—The Corliss gear on account of this action is sometimes called a *releasing or drop* gear.

The action of the releasing gear being somewhat complicated, will now be explained in detail.

The Releasing Gear.—This part of the Corliss valve gear, as designed by the different builders, varies in detail, but not in principle. There are, however, two general classes of releasing gears:

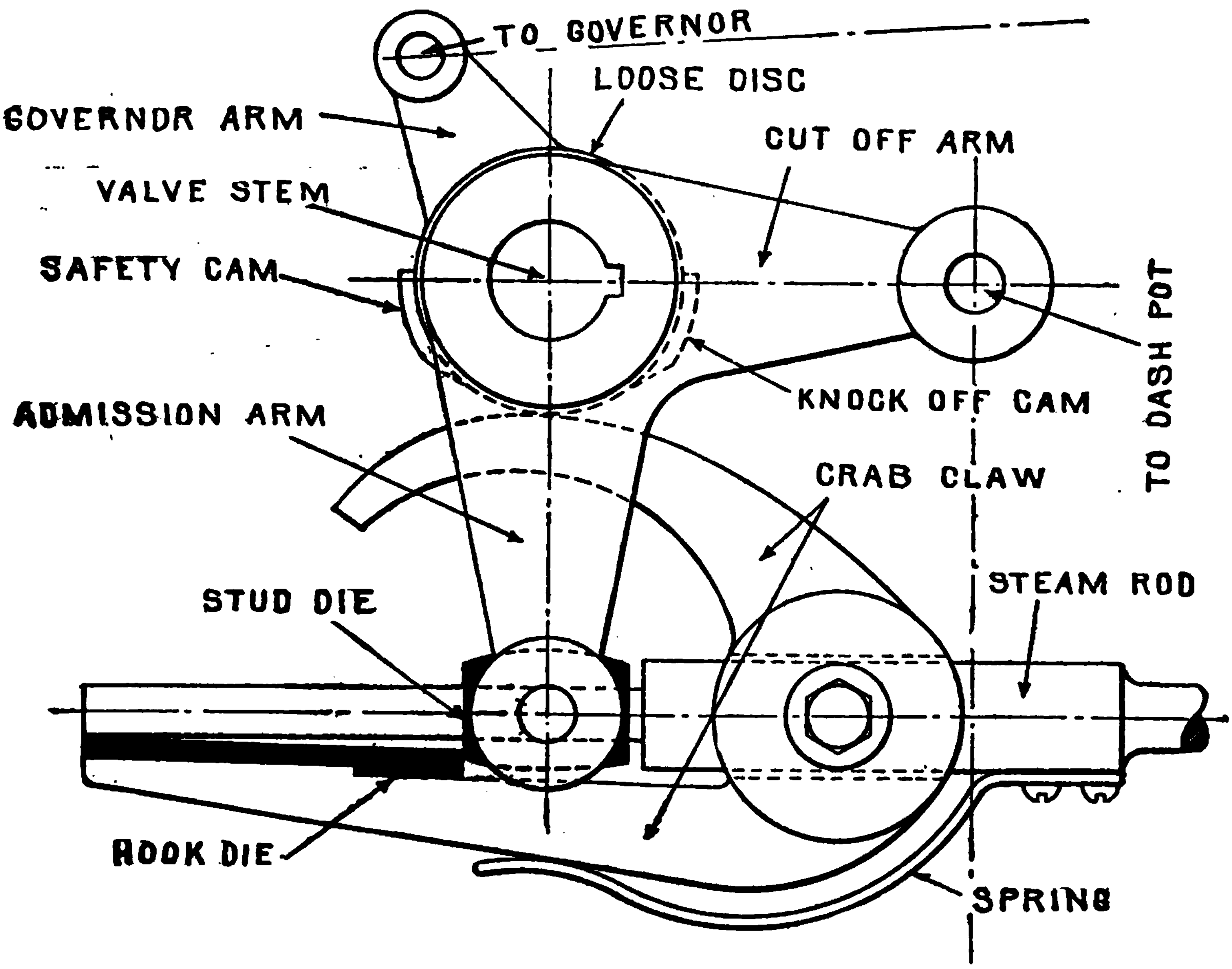


FIG. 900.—The crab claw releasing gear; view showing parts. *In operation,* the admission arm is moved by the steam rod to the right giving outside admission. When in its movement the crab claw is depressed by the knock off cam, the hook die releases the stud die, and the valve is suddenly closed by the dash pot thus cutting off steam.

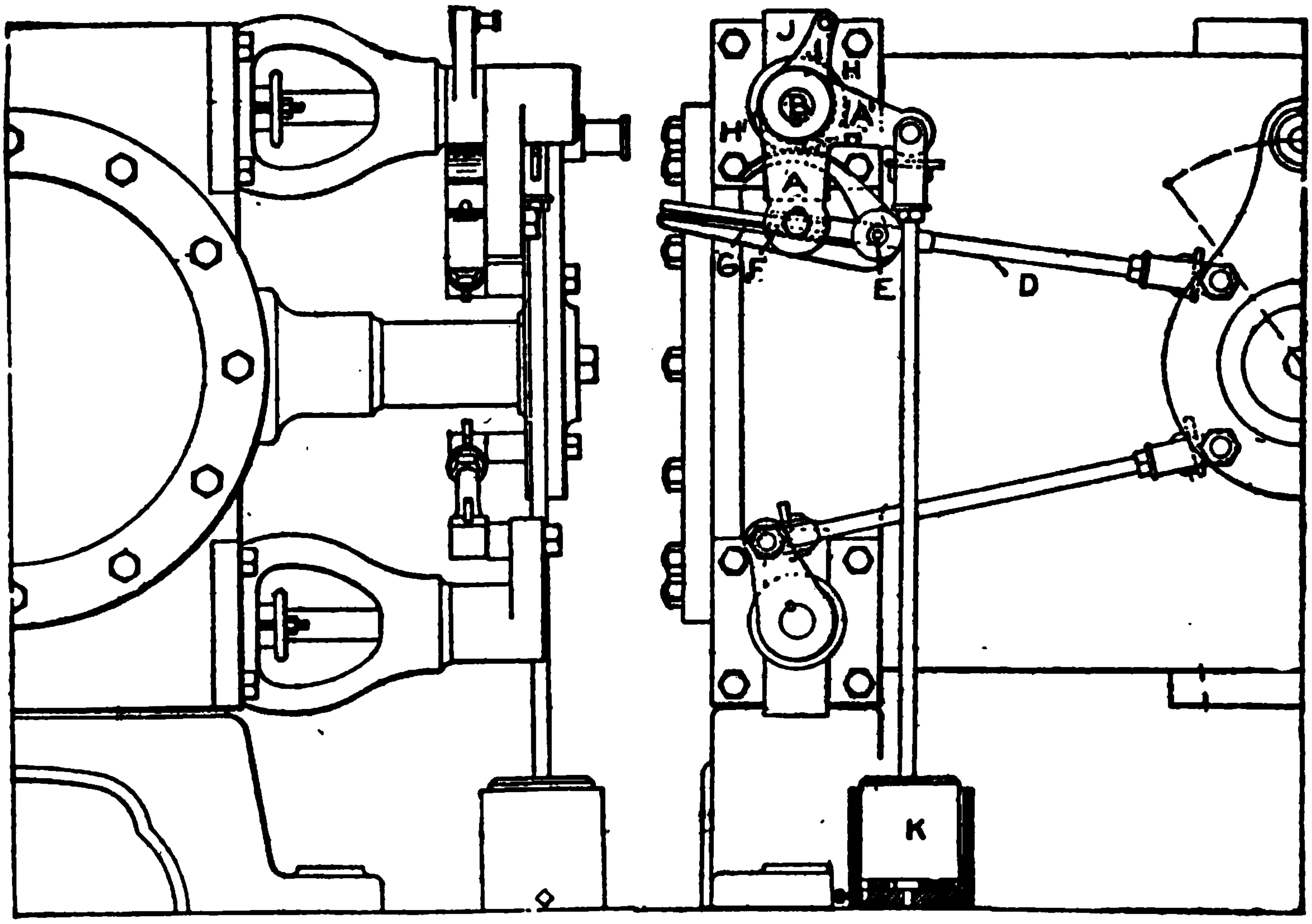
1. Those in which the valves rotate toward the center of the cylinder in admitting steam, as in fig. 897.

2. Those in which the valves rotate toward the ends during admission, as in fig. 898.

Among the gears included under the first division are the familiar *crab claw*, and *half moon* types; those of the second division are generally of a form known as the *oval arm gear*.

Crab Claw Gear.—This type of releasing gear was used on the original Geo. H. Corliss engine, and is still to be found in operation.

As shown in figs. 901 and 902, the valve stem passes through a stuffing box, and a bracket which forms a bearing. Keyed to the end of the valve



FIGS. 901 and 902.—The crab claw releasing gear as used on the Geo. H. Corliss engine. This type of gear is still in use.

stem is a crank having two arms A, and A,'* the arm A, is for admission, and A', for cut off.

The *admission arm* terminates in a bearing, through which passes the *steam rod* D; this bearing contains a *stud die* F. There is a *disc* which turns loosely on the valve stem between the crank and the bracket bearing;

*NOTE.—On the half moon and oval arm gears, there is only a single crank keyed to the valved stem; it is called the *steam arm*.

a projecting arm I, connects this with the ***governor rod***. Attached to the circumference of the disc is a ***knock off cam*** or ***button*** H, and a ***safety cam*** or ***button*** H'. These cams are shown more clearly in fig. 900.

A "***crab claw***" is pivoted to the steam rod at E, and pressed against the disc by a ***spring***. Attached to the lower member of the crab claw is a ***hook die*** G.

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FIG. 903.—Sectional end view of Providence Corliss cylinder showing valve gear and placement of engine on foundation.

An important part of the releasing gear is the safety cam H′; its object is to prevent the engine running away in case the governor belt break. This cam, as shown, is attached to the governor disc at such a point that when the latter is in its extreme position (in the direction of late cut off), the cam will depress or "knock off" the crab claw, and thus prevent engagement of the hook and stud dies, resulting in the valve remaining closed.

If the governor belt should break while the engine is running, the governor balls would fall to their lowest position, and thus turn the knock off lever to the extreme position which will bring the safety cam into engagement with the crab claw. The hook then will not raise the steam arm, and the valve will remain closed, thus shutting off the steam supply.

To avoid disaster in case the governor belt break, the engineer should not fail to put the governor safety stop in "running position,"

FIG. 904.—Section of Rice and Sargent inlet steam bonnet and valve stem, showing stuffing box, drain pipe, etc.

after starting the engine, otherwise the safety cam will not operate.

Attached to the *steam arm* is a rod connecting with the *dash pot*, which, by its action, as soon as the dies release, quickly closes the valve. The dies are released by the left hand end of the hook striking against the knock off cam, thus causing the supply of steam to be cut off.

The knock off cam is a part of the knock off lever, and the latter is connected with the governor by the *governor cam rod*. The point of cut off, therefore, is determined by the position of the knock off cam which is controlled by the action of the governor.

KNOCK OFF

FIGS. 905 and 906.—The half moon releasing gear. A type extensively used. There is only one arm attached to the valve stem, the cut off or hook arm which forms a part of the bell crank of half moon is free to revolve on the stem. The various parts of the gear are named in the figures.

Half Moon Gear.—This type of releasing gear is illustrated in figs. 905 and 906. A single crank, or *steam arm* is keyed to the valve stem, motion being transmitted to the valve by this arm, both for admission and cut off. The valve stem passes through a stuffing box and its outer extremity is supported by a bearing in the bracket.

At the cut off end of the *half moon*, is a stud upon which is pivoted the hook arm.

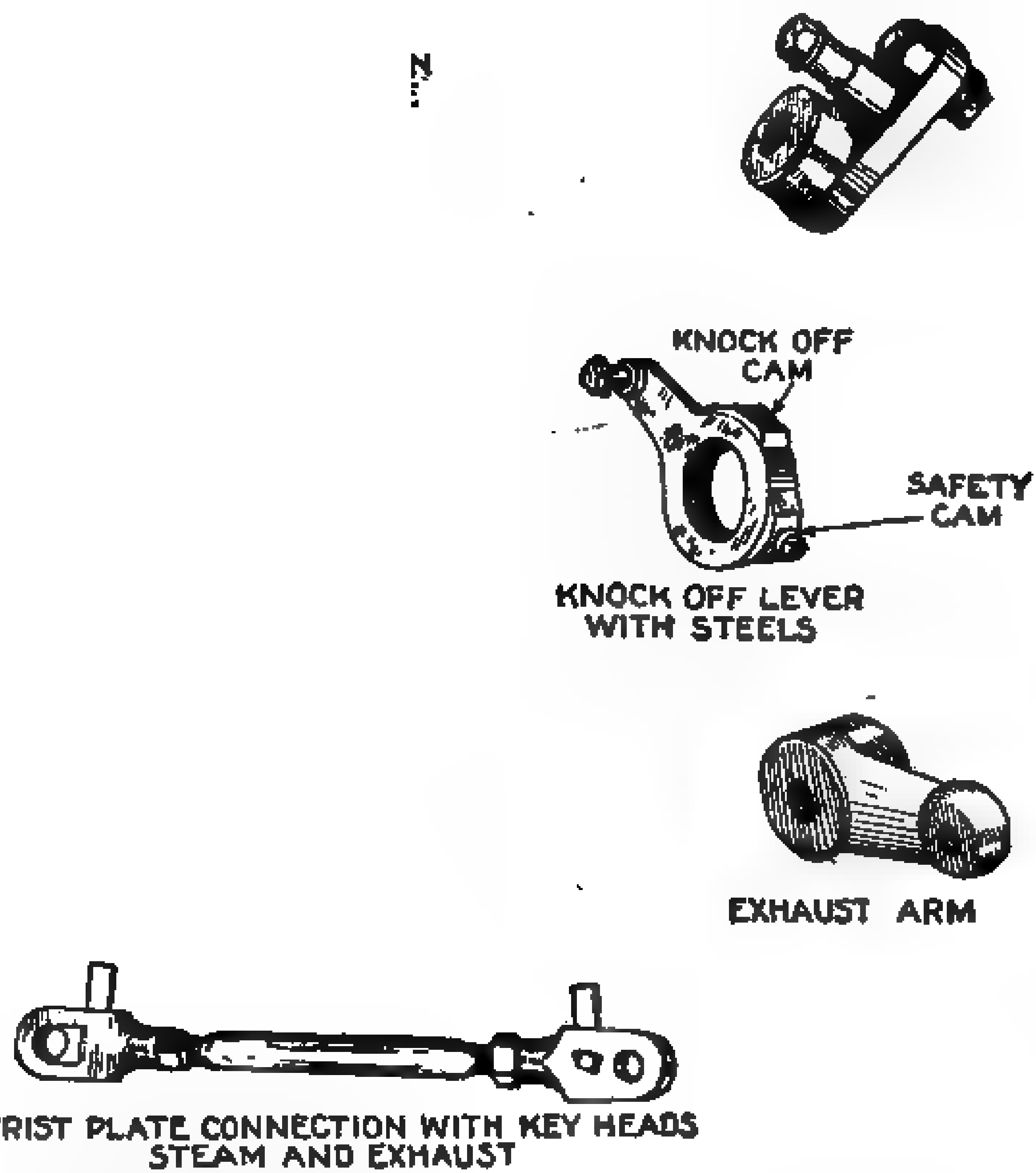
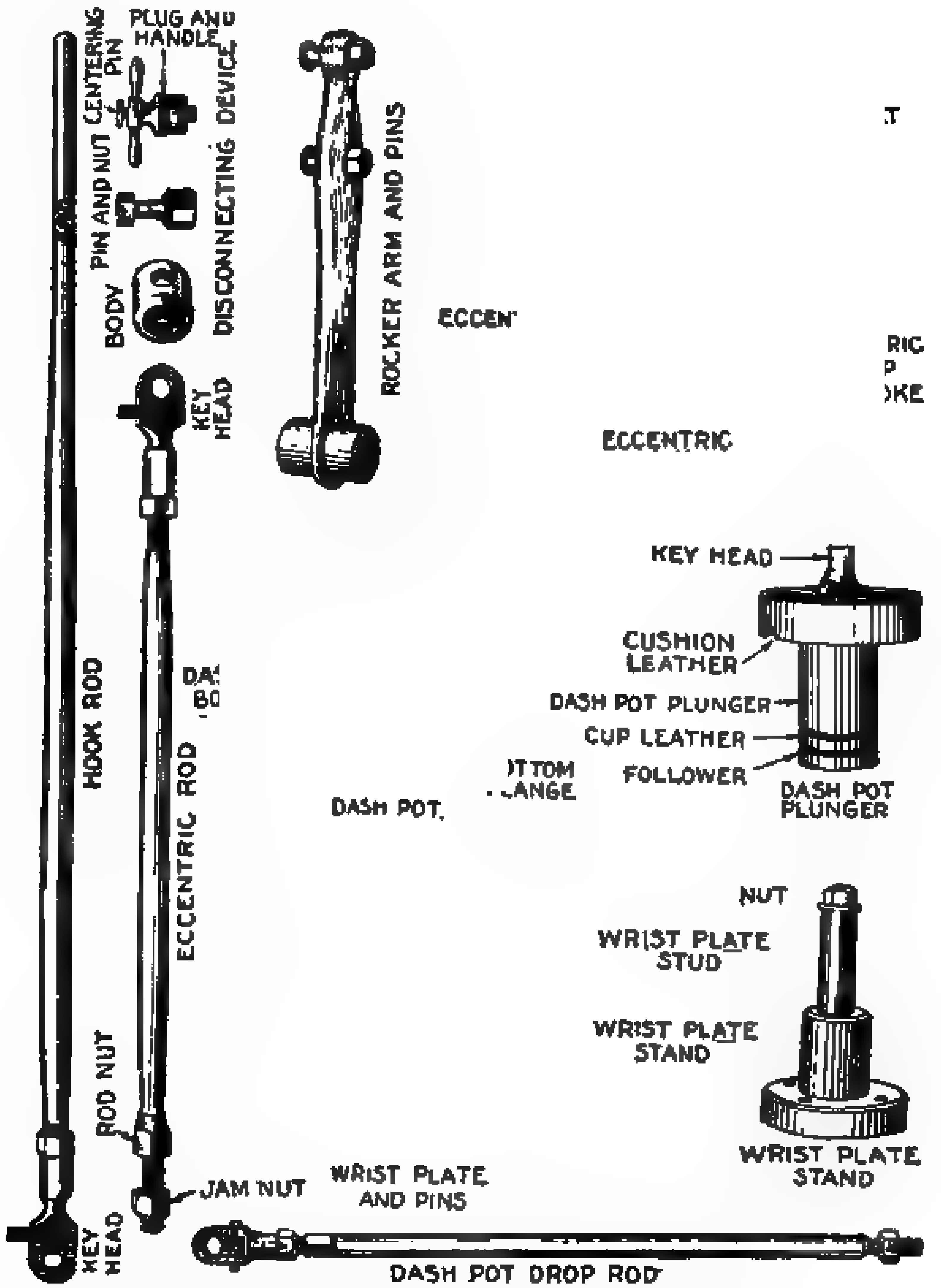


FIG. 907 to 914.—Allis-Chalmers oval arm releasing gear. Views showing the mechanism assembled, and taken apart.



FIGS. 915 to 926.—Views showing connections and parts between the wrist plate and shaft, also the wrist plate, and dash pot of Allis-Chalmers oval arm gear.

A hard steel block, or die, projects at the extremity of this arm, forming a *hook*. Another block called the *stud die*, is attached to the steam arm. A spring presses the hook arm toward the stud die.

The *governor disc*, which has an arm, or knock off lever, works loosely on the valve stem, and has attached to it the *knock off* and *safety cams*. The hook arm has a branch called the *tail arm* projecting toward the governor disc, there being a *button* fastened to the end for engagement with the cams. A rod connecting with the dash pot plunger is pivoted to the steam arm about half way between the hub and stud die.

FIG. 927.—The Murray-Corliss releasing gear. An example of the oval arm gear which is used with inside admission valves.

When the steam rod moves in the direction for admission, the half moon will rotate and raise the hook stud.

As the hook arm rises, the die engages the stud die thus lifting the steam arm.

During the movement, the hook stud moves in an arc about the valve stem, and at a point determined by the governor, the button on the tail arm strikes the knock off cam which knocks the hook die away from the stud die, allowing the valve to drop shut by the action of the dash pot.

A safety cam is provided which engages with the button when the governor is in its lowest position, thus holding the dies apart in case the governor belt break.

Oval Arm Gear.—This is practically the same as the half moon gear, with the exception that the admission arm as shown in figs. 927 and 928 projects upward in order to get the opposite

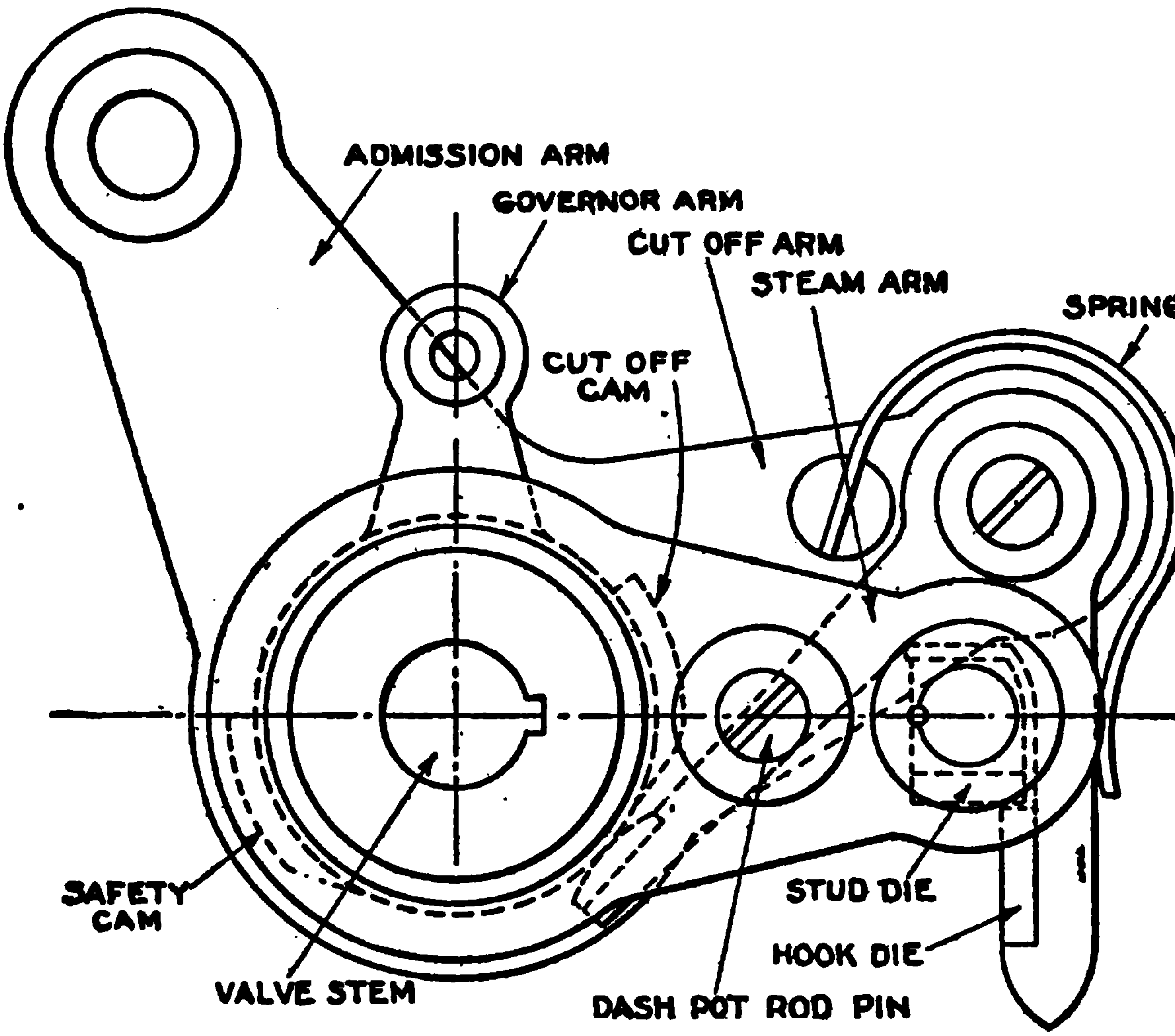


FIG. 928.—Detail of oval arm gear.

motion of the valve for outside admission. The similarity of the two gears is seen from the illustrations.

Dash Pots.—The quick closing of the steam valve giving a sharp cut off, is one of the excellent features of the Corliss gear. In the earlier engines, the steam valves were snapped shut by springs, or weighted plungers, but later, a more satisfactory

operation has been obtained by means of *dash pots*, whose operation depends on:

1. *A partial vacuum*, to secure quick closure of the valves;

2. *An air cushion*, to absorb the momentum of the moving parts, bringing them quietly to rest.

To obtain this pneumatic action, the dash pot is fitted with an air tight plunger. A dash pot is simply *a cylinder closed at one end, and accurately bored to receive the plunger*. A rod connects the latter to the steam arm of the valve. The steam

FIGS. 929 and 930.—Rice and Sargent dash pots. Fig. 929, vacuum type; fig. 930, spring type. According to the makers, vacuum pots are preferably limited to speeds of 125 revolutions per minute, and spring pots to 250 revolutions. Their strengths on normal lifts are the same. On light loads where a vacuum pot of the best design may not close properly, the spring pot will give results equally good as with a heavier load. The dash pots of either type are placed with their top flush with the sole plate, and no piping connections are necessary. Control valves with easily accessible handles are provided for adjustment. The air is confined within the passages of the pot and so gives practically soundless operation.

arm, then, as it is raised by the hook, lifts the plunger which produces a partial vacuum in the dash pot. When the hook releases the steam arm, the pressure of the atmosphere on top the plunger causes it to quickly drop and close the valve. The compression of air remaining below the plunger forms a cushion which prevents shock.

The pressure in this type of dash pot, then, must be below that of the atmosphere during the first portion of the downward stroke in order to

shut the valve, and greater during the latter portion, to bring the parts to rest. From this it follows that the pull on the dash pot rod is greatest at the beginning of the downward stroke of the plunger, and on account of the low initial vacuum, it is very little if any at the point where the steam valve closes. The speed of the valve at this point, therefore, is not as great as it might be if the effect of a good vacuum were available during the entire period of closure of the steam valve.

In order to obtain this action, the single plunger dash pot, as just described, has been modified by the addition of another plunger or piston. Here the compression is carried on in a separate cylinder, which permits of a higher vacuum for closing the valve.

FIG. 931.—The Harris-Corliss dash pot with separate compression cylinder. The parts are: V, vacuum cylinder; C, compression cylinder; P, vacuum plunger; P′, compression piston; K, check valve; M, port to atmosphere; S, compression regulating screw.

Separate compression dash pots are used on most Corliss engines and a typical construction is shown in fig. 931.

As shown, the dash pot consists of a vacuum cylinder V, and a compression cylinder C. Fitted to these are a plunger P, and piston P′, forming one casting. At the end of the vacuum chamber is a sensitive check valve K; there is a passage M, leading

from the compression chamber to the atmosphere, and its opening is regulated by a screw valve S.

The operation is as follows:—As the steam arm rises, the dash pot rod lifts the plunger P, causing the check valve S, to close, producing a high vacuum in the vacuum cylinder V. The travel of the plunger being longer than the compression cylinder C, the piston P, is carried past the end of this cylinder, thus giving access to the atmosphere.

When the steam arm is released, the high vacuum in the cylinder causes the plunger to drop with great speed until the piston enters the compression cylinder. The air contained therein is then compressed which quietly brings

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the moving parts to rest. The amount of cushion thus obtained may be regulated by the screw valve S, allowing more or less air to escape during the compression.

Corliss Engine Speed.—One inherent defect of the Corliss engine is that the releasing gear is not adapted to high rotative

speed. In many power installations, this is a drawback, not only because of the increased first cost, but on account of the space occupied, and in general because higher speeds are attended with greater saving in the use of steam.*

For many years 70 to 80 revolutions per minute for the small and medium sizes was considered the safe limit of rotative speed, and later from 90 to 100. Since then Corliss engines are commonly run at 100 to 120, and those especially built for high speeds, as high as 200 revolutions.

FIG. 933.—The Frick-Corliss dash pot. A single cylinder type in which both vacuum and compression are obtained in one cylinder. The regulating device is shown at the right. This pot necessarily operates on a vacuum of lower degree than the separate compression type, hence, a larger plunger is used to obtain the required pull.

Various refinements have been made in the design of the releasing gear to secure satisfactory working at high speed.

*NOTE.—According to tests made by Prof. Jacobus, on a 17×30 engine, fixed cut off, Meyer valve, the loss in economy for about one-fourth cut off is at the rate of one-twelfth lb. of water per horse power for each decrease of a revolution per minute from 86 to 26 revolutions, and at the rate of five-eighths lb. of water below 26 revolutions. It should be noted, that the results thus obtained do not represent the *absolute* loss, because, an engine designed for slow speed would have correspondingly small ports and passages thus reducing the clearance, which would considerably reduce the loss above obtained.

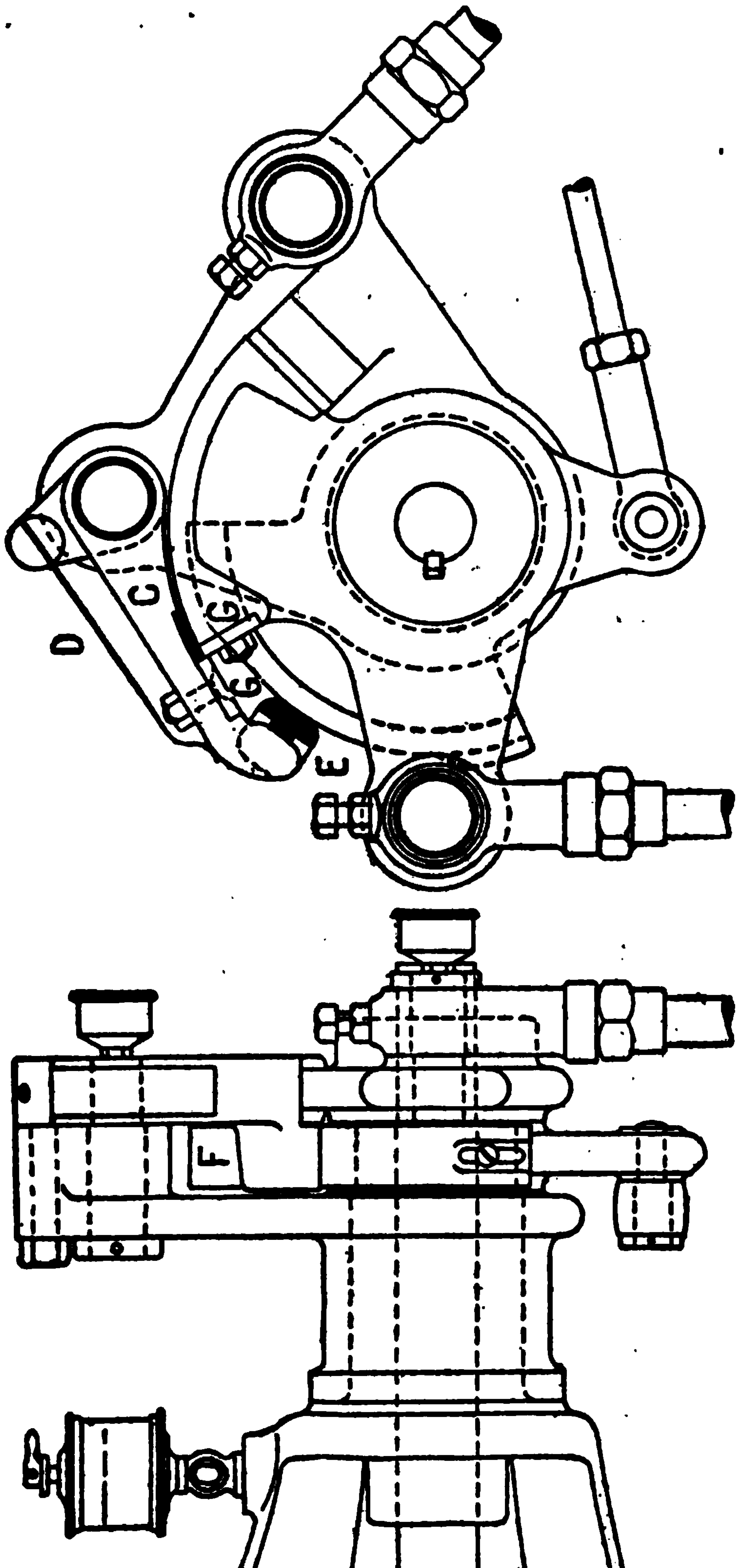
The inertia of the releasing parts has been reduced as much as possible by making them of minimum weight, thus rendering the dash pot action more effective.

Larger dash pots have been employed to secure quicker action in closing the valves.

Fig. 934.—Wisconsin-Corliss high speed releasing gear. The features of this gear are the outside bearing for the valve stem, short leverage, and large dies; these are desirable qualities for high rotative speed.

Fig. 935.—Wisconsin-Corliss high speed releasing gear. View from cylinder showing T end of valve stem, and the releasing features.

The motion of the gear has been reduced by shorter leverage and double ports.



FIGS. 936 and 937.—The Franklin high speed releasing gear as applied to the Hewes and Phillips-Corliss engine. The latch C, has attached a fibre block E, pressed against the cut off cam F, by gravity and the spring D. When the block is on the lower portion of the cam, the dies G, G, engage for admission; when E, rides on the high part of the cam, the dies release for cut off.

These several refinements permit considerably higher rotative speeds, than is permissible with the ordinary construction.

An example of a high speed Corliss valve gear designed as above outlined is shown in figs. 934 and 935.

Besides the lightness of parts, short leverage, and reduced motion, the steam arm is located inside the bracket between the stuffing box and bearing. The stem is thus supported on both sides of the steam lever instead of having the latter overhung. This gear, according to the builders is designed for speeds up to 175 revolutions per minute.

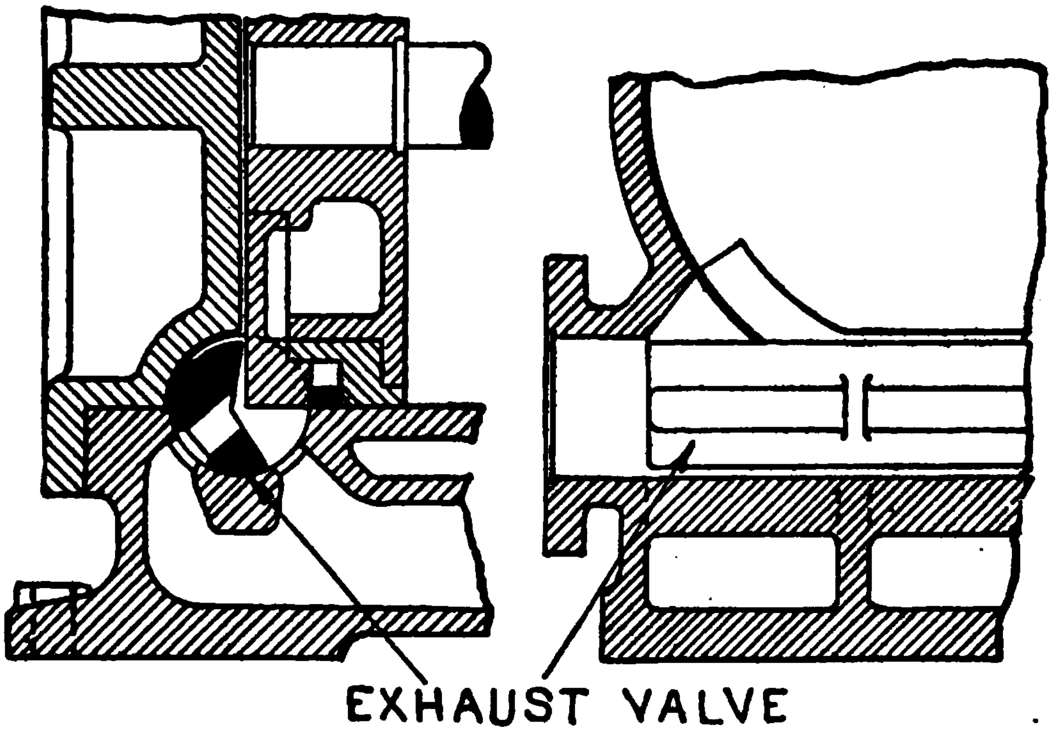
In figs. 936 and 937 is shown

the Franklyn gravity releasing gear for which is claimed satisfactory working up to 200 revolutions per minute.

Its chief feature is the placing of the hook arm in such a position that gravity assists the spring in moving the arm to engage the stud die of the steam arm.

Ques. In what respect is the Corliss gear limited when operated by a single eccentric?

Ans. *The releasing mechanism will not act to cut off later than about one half stroke.*



FIGS. 938 and 939.—Wisconsin-Corliss small clearance exhaust valve. The clearance is reduced by locating the valve so that its axis is tangent to the cylinder bore; the valve is thus contained partly in the cylinder casting and partly in the cylinder head.

Ques. Why is this?

Ans. If the eccentric be set at 90° ahead of the piston, it will reach its extreme position when the crank arrives at half stroke. If at this point, the hook, which was rising and opening the valve, has not struck the knock off cam, it cannot strike at all, since the motion of the wrist plate is reversed at this point; hence, the hook and steam arm will begin to descend thus gradually closing the valve. This will take place near the end

of the stroke but it will not be the sharp cut off as produced by the sudden drop of the dash pot plunger when the dies are released.

Ques. Why is the single eccentric gear unsatisfactory for a cut off later than one half stroke?

Ans. In order to cut off later than half stroke, the eccentric must be turned back until it is less than 90° ahead of the crank. This decrease in the angular advance causes release and compression to occur too late. If exhaust lap be added to correct one, it will increase the error of the other in like proportion.

For instance, if the exhaust valves be given negative inside lap to obtain earlier release, compression will be delayed an equal amount.

The Use of Double Eccentrics.—In the preceding questions, it is seen that the Corliss gear with one eccentric is limited to cut offs not exceeding half stroke.

If a single eccentric be set with the least possible angular advance, in order to get the maximum range of cut off, the engine when heavily loaded will not be able to exhaust the steam early enough to sufficiently reduce the pressure at the beginning of the return stroke.

If the eccentric be advanced to secure early release, the range of cut off available with the releasing gear is so reduced that the knock off cam may not strike the hook for one or two revolutions thus increasing the trouble which it was desired to remedy.

The limitations of the single eccentric gear are overcome by the addition of a second eccentric; *one to operate the steam valves, and the other for the exhaust valves*. Admission and cut off are thus made independent of release and compression. The steam eccentric may therefore be set with *negative* angular advance to obtain a late cut off, and the exhaust eccentric with *positive* angular advance to secure early release and compression.

FIGS. 940 and 941.—Vilter cylinders with *single* and *double eccentrics*; fig. 940, single eccentric valve gear; fig. 941, double eccentric valve gear. For ordinary power purposes the single eccentric is satisfactory, and with it a maximum cut off of one-third is obtainable. Cut offs greater than this are not practicable owing to the fact that compression and release are reduced to such an extent that pounding of the engine will result. This is due principally to the fact that the functions of the exhaust valve are interfered with. For simple non-condensing single eccentric engines, the usual point of cut off is about ¼ stroke, the same being true of simple condensing engines. This cut off insures proper compression at all speeds. *Double eccentric engines,* in which the exhaust valves are actuated by a separate eccentric independently of the steam valves, permit of almost any range of cut off without interference with the proper functioning of the exhaust valves. Where the load conditions are of a fluctuating character, and where heavy over loads frequently occur, as in the driving of electrical machinery in connection with lighting and railway service, double eccentrics are essential, as they permit of any range of cut off up to 80 per cent. without affecting the compression.

It should not be inferred from the preceding paragraph that a single eccentric gear, with the addition of another eccentric, will give as good results for long range cut off as if it were originally designed for double eccentrics. The reason for this is because in order to get a long range cut off, the steam eccentric must be given considerable *negative* angular advance, and the more the eccentric is set back the slower is the movement of the valve in opening the port

to delay the closing of the valve until after release in which case live steam would blow through the cylinder into the exhaust pipe. To guard against this, valve gears when designed for two eccentrics have a maximum cut off of almost full stroke within the range of the releasing gear. This insures the release of the steam arm for any load within the capacity of the engine.

Since the necessary negative angular advance which must be given the eccentric to secure a very late cut off, reduces the speed at which the valve opens the port, poor admission is avoided in designing a double eccentric gear, either by increasing the port opening, or by the use of double ports.

FIG. 943.—The Corliss double eccentric valve gear. The admission, and exhaust valves are operated independently from separate wrist plates S and E. A long range cut off is possible with this gear because the steam features may be varied at will without disturbing the exhaust.

An example of a double eccentric valve gear is shown in fig. 943. There are two wrist plates, which work independently of each other. The steam valves are connected to the upper plate S, which is operated by the steam eccentric, and the exhaust valves are connected to the lower plate E, operated by the

Corliss engine and is well adapted for light duty.

Figs. 944 and 945 show a front and rear view of a girder frame; a heavy backbone projects from the rear, as illustrated in fig. 945, directly in line with the strain. The form of the girder frame is such that the required strength and stiffness is secured with the least amount of metal

Where the conditions of operation are more severe, a frame of heavier construction is desirable, such as shown in figs. 946 and 947.

Here the metal extends down to the foundation; the frame is hollow and of rectangular cross section, forming a kind of box girder of the so

FIGS. 946 and 947.—Front and rear views of a semi-heavy duty Corliss frame. The metal is carried down to the foundation, giving a more substantial construction.

called "straight line" design in which the walls extend in practically straight lines from the main bearing to the cylinder, this bringing the metal, as near as possible, in the direction of the strain so as to reduce bending, or twisting stresses.

With the advent of electric railways, and the plan of transmitting power by electricity, a new type of frame came into use, which has become generally known as the "heavy duty," or Tangye. The true Tangye frame was designed by an

English engineer of that name,* and is to be found on Corliss engines only in its modified form, such as is shown in fig. 948.

Starting with a massive bearing, the metal is placed in straight lines between it and the guide casting to which it is bolted, although in the smaller sizes the frame proper and guide are sometimes cast in one piece.

The frame terminates in lines of symmetry and strength to form a circular end which receives the guide. The graceful down sweep curve at this point is characteristic of the Tangye design.

The end and side walls are carried down, and sometimes an enclosing bottom is provided, which distributes the weight and stresses to the foundation, and also forms a receptacle to catch oil.

FIG. 948.—Vilter heavy duty Corliss engine.

FIG. 949.—The Tangye, or heavy duty frame. The graceful down sweep curve is characteristic of this type. The original design is in one piece without the separate guide casting as on the Porter-Allen engine.

*NOTE.—The regular Tangye frame is in one piece with the down sweep curve beginning at the cylinder; it is adapted to short stroke engines having locomotive guides, such as the Porter-Allen, Buckeye, etc.

To meet the severest conditions of service where high speed and high pressure are required, a frame of the most substantial construction is necessary. This type is known by some as the *rolling mill frame*, because of its universal adoption for that service.

An example of an extra heavy duty frame is shown in fig. 950; it is similar to the Tangye frame illustrated in the preceding figure, but of more massive

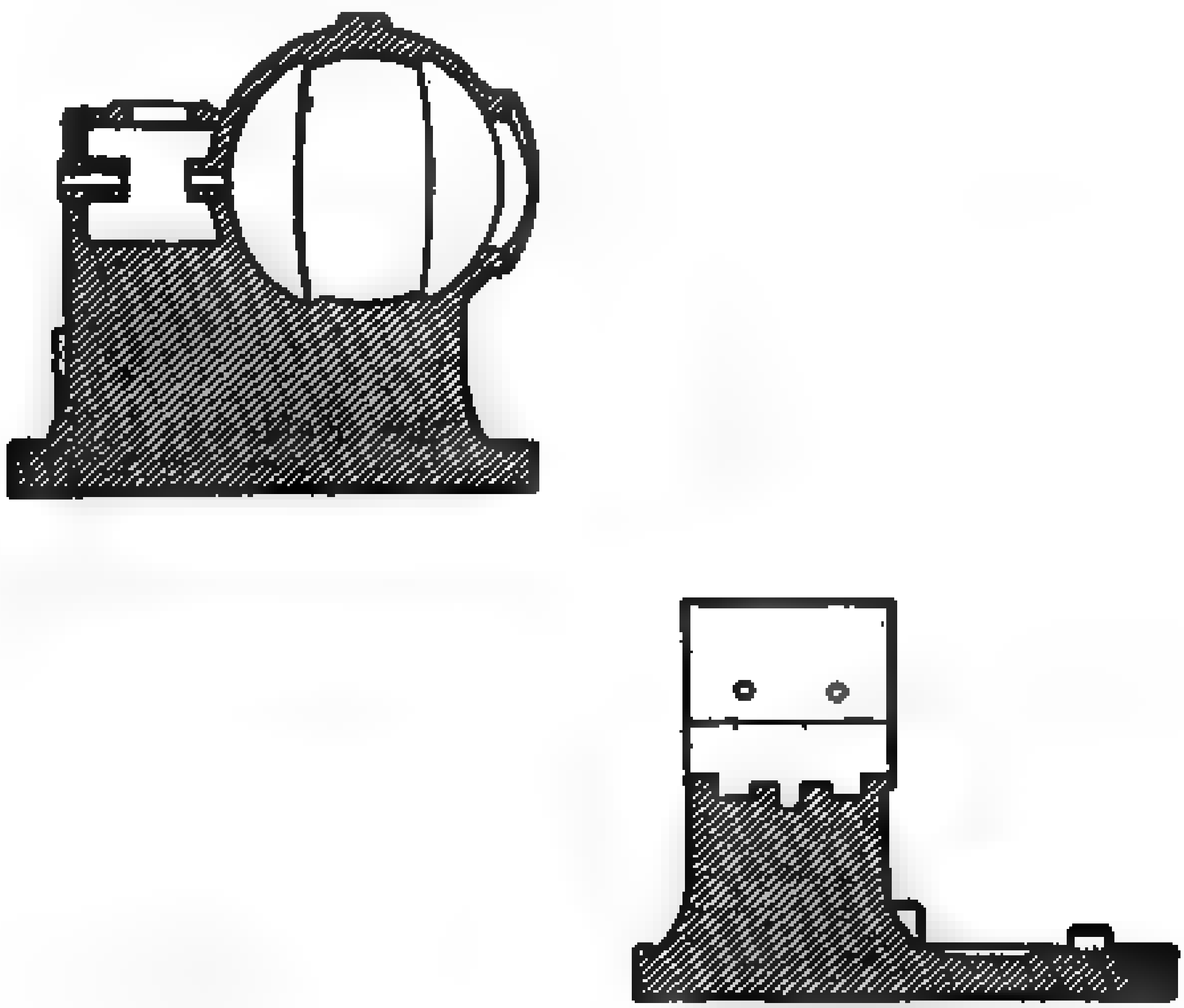
FIG. 950.—A rolling mill, or extra heavy duty frame; this is a modified Tangye frame of very substantial construction.

FIGS. 951 and 952.—Two views of the Murray-Corliss rolling mill frame. A design adapted to the most severe conditions of engine operation.

construction. The frame and guide are cast in one piece, and it should be noted that the metal of the latter is carried down to the foundation giving additional support.

Another design of extra heavy duty frame is shown in figs. 951 and 952. This is also a one piece frame, and its very substantial and massive construction is indicated by the cross sections, figs. 953 to 956.

Setting Corliss Valves; Single Eccentric.—Adjusting the Corliss valve gear should present no difficulty when once its construction and principles of operation are understood. In



FIGS. 953 and 956.—Cross sections of the Murray-Corliss rolling mill frame, showing the heavy and substantial construction.

fact, part of the work has been done by the engine builders in scribing the necessary reference marks on the gear during its construction. Marks corresponding to the steam edges of the valves and ports are scribed on the end of each valve and seat respectively, as shown in figs. 877 and 878.

There are on the back of the wrist plate hub three marks, A, B, C, as shown in fig. 957, and one, D, on the wrist plate stud.

When A, registers with D, the wrist plate is in its neutral position; similarly when B, or C, registers with D, the valve is in one of its extreme positions. In some cases the marks are arranged as shown in fig. 960.

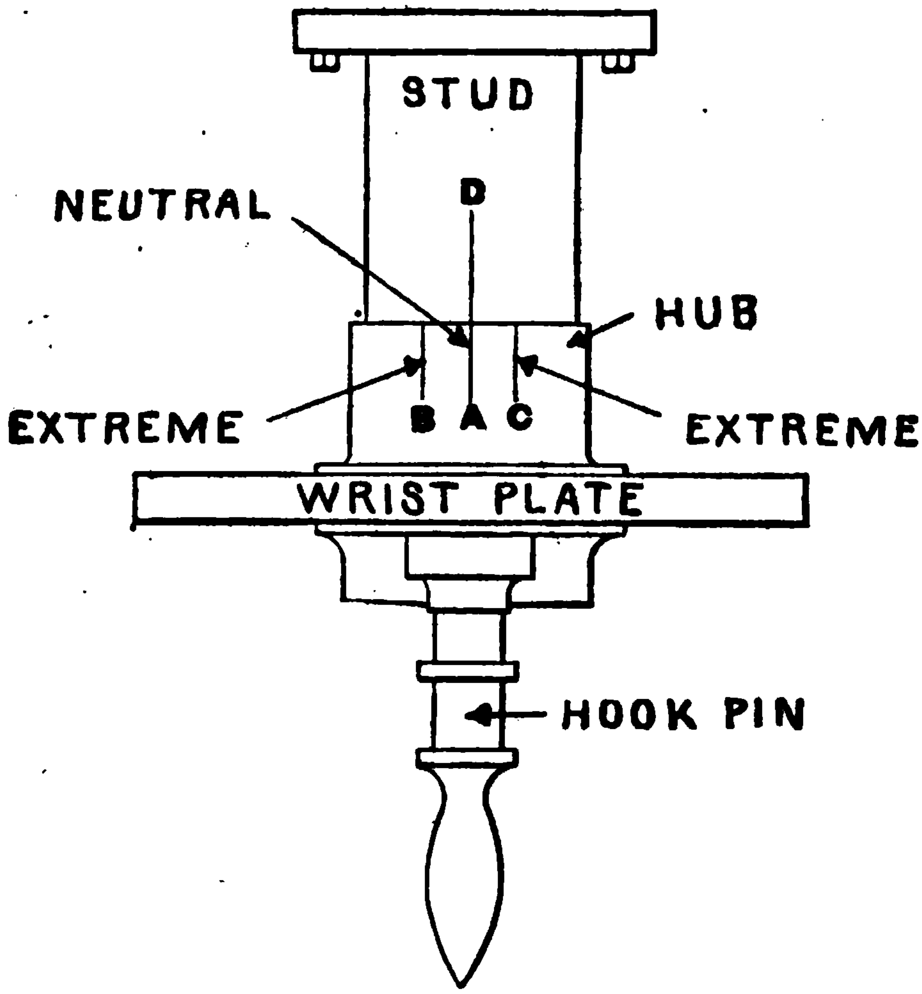


FIG. 957.—Reference marks on the wrist plate and stud for valve setting. When D, registers with A, the wrist plate is in its neutral position; B and C, indicate the extreme positions.

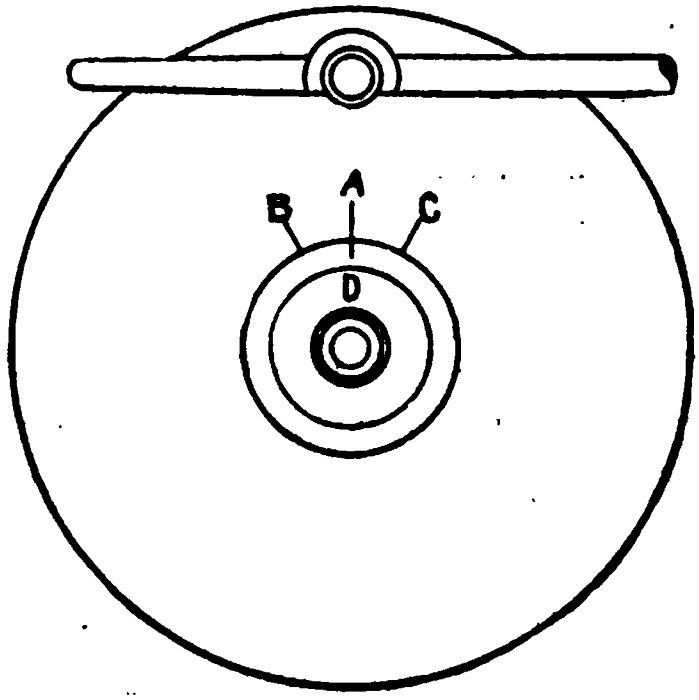


FIG. 958.—A second arrangement of the wrist plate and stud reference marks. In this case the marks are reversed, A, B, and C, being on the plate, and D, on the stud.

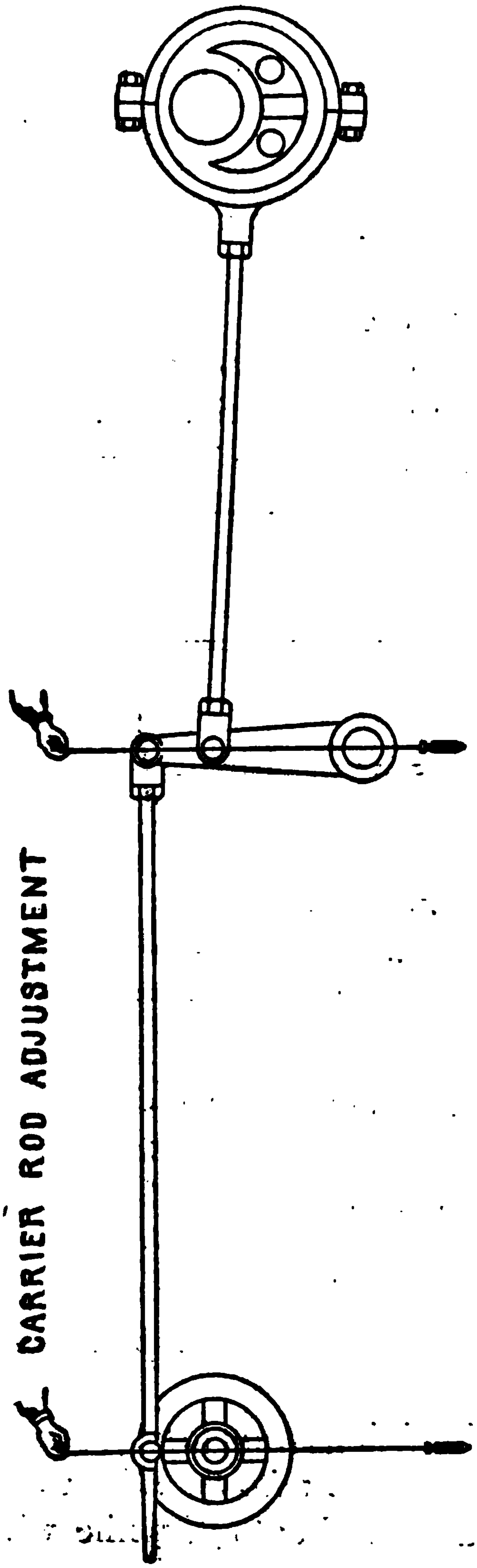


FIG. 959.—*Squaring the wrist plate and rocker;* the use of plumb bobs, to accurately and conveniently perform this operation, is here shown.

The several operations to be performed in setting Corliss valves are as follows:

1. *Squaring the wrist plate and rocker;*
2. *Squaring the valves;*
3. *Adjusting the dash pot rods;*
4. *Setting the eccentric, equalizing the lead;*
5. *Adjusting the governor connections.*

1. Squaring the Wrist Plate and Rocker.—The first step in setting the valves is to unhook the carrier rod, and put the wrist plate in its neutral position, that is, the position shown in figs. 957 and 960, where line A, is opposite D.

The wrist plate should be clamped in this position by placing a piece of paper between it and the washer on the supporting pin. If the marks A, and D, have been correctly located, a plumb line from the center of the hook pin as shown in fig. 959 should register with the center of the wrist plate.

The rocker should now be placed in a vertical position with

a plumb bob, and the length of the carrier rod adjusted, if necessary, so that it will engage with the hook as shown in the figure.

The rocker is conveniently brought into a vertical position by turning the engine over until it is vertical as determined by the plumb bob, taking special care *before* doing this *to shorten the dash pot rods enough to prevent the steam arm die being forced against the hook arm shoulder.*

2. Squaring the Valves.—When the back bonnets are removed, the reference lines which indicate the steam edges of the valves and ports will be visible as shown in fig. 960. The distance between these lines P, V, will indicate the lap when the

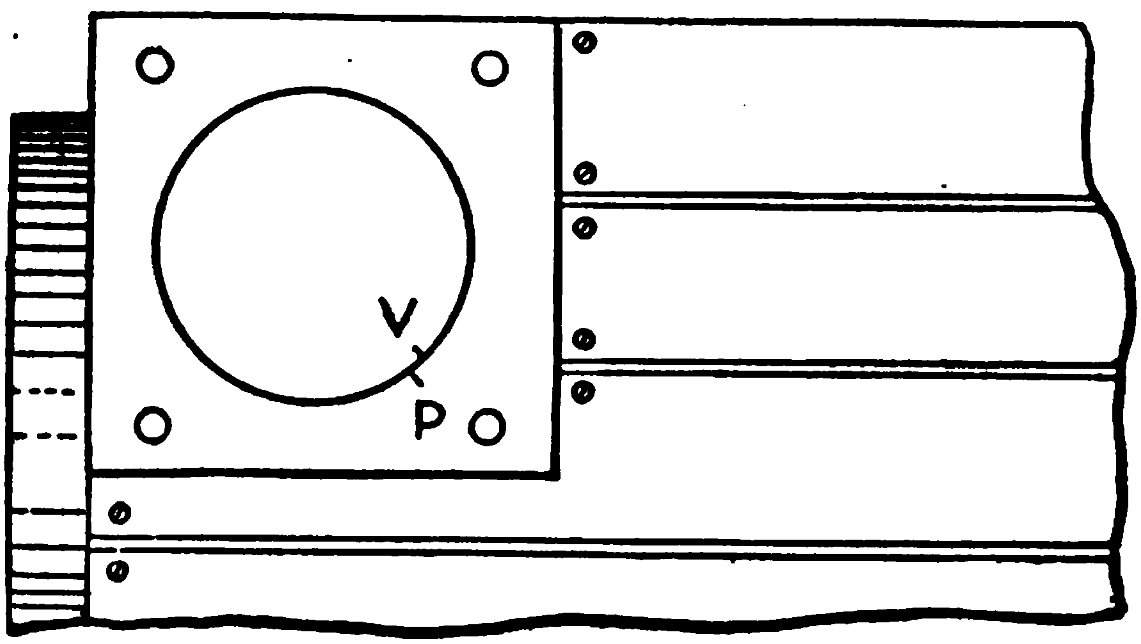


FIG. 960.—Valve and port reference lines. When the bonnet is removed these lines are visible, and are used to set the valve.

wrist plate is in its neutral position. Being thus guided, the steam valves are given a slight amount of lap by lengthening or shortening the steam rods K, K, fig. 961, the wrist plate being still clamped in its neutral position.

Similarly, the exhaust valves are put in *line and line* position or given a very small amount of lap by adjusting the exhaust rods L, L.

It should be noted that the **object** in giving *steam lap* to Corliss valves is quite different *from the result sought with the slide valve*: Lap is given a slide valve *to obtain a desired cut off*, while with

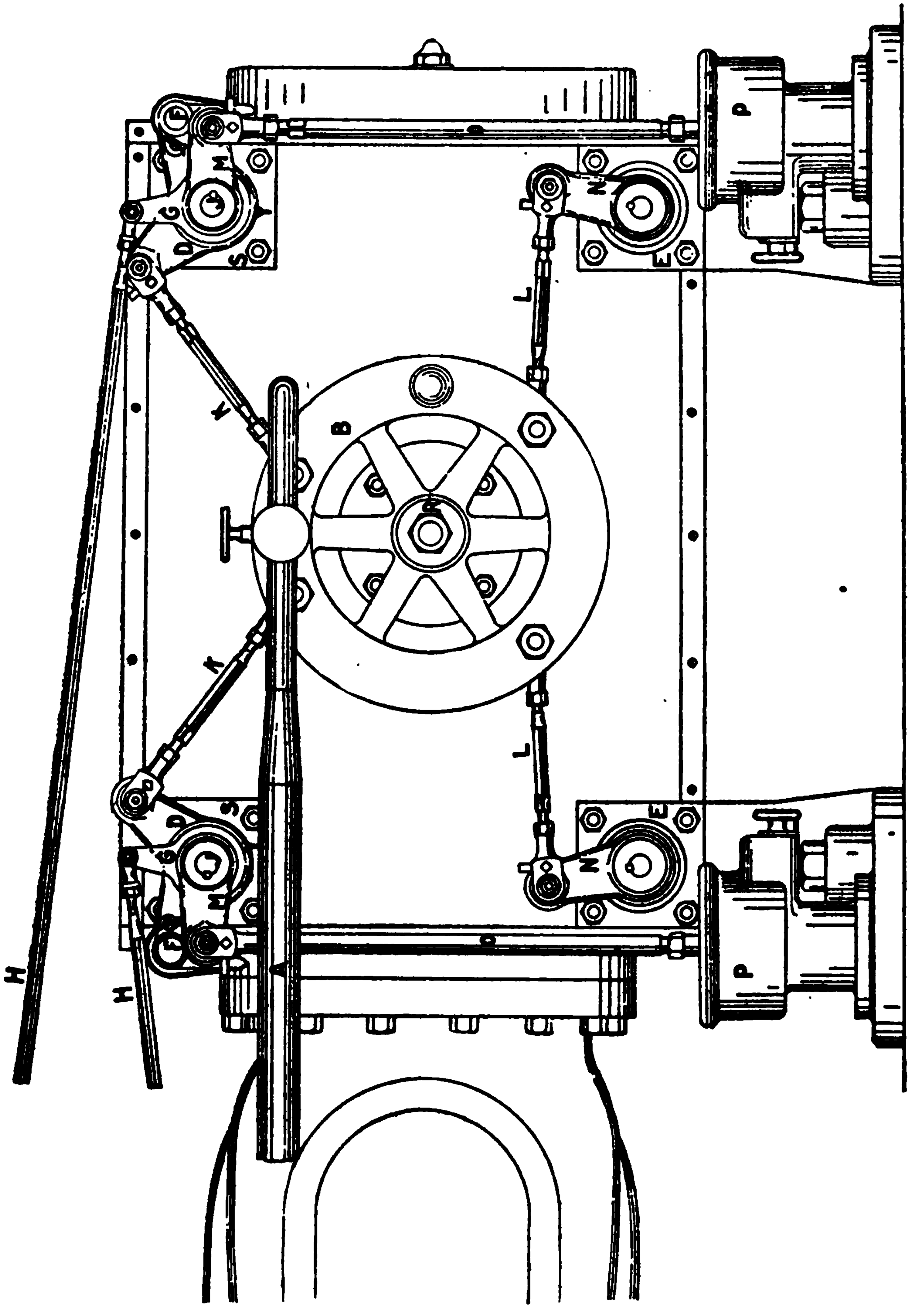


FIG. 961.—*Squaring the valves.* The wrist plate B, is clamped in neutral position, and the steam valves given the proper lap by adjusting the steam rods K, K; similarly, the exhaust valves are placed in their lap, or neutral position by adjusting the exhaust rods L, L.

the Corliss gear, its object is *to secure the most favorable angular advance of the eccentric.*

As has been mentioned, the eccentric should have some angular advance in order to secure pre-release, and the proper amount of compression.

The usual amount of steam lap for small engines is from 1/16 to 1/4 inch, and from 1/4 to 1/2 inch for the larger sizes.* The builders of the Reynolds-Corliss engines recommend that the valves be set according to the following table lap and lead.

Table for Setting Valves

Diameter of Cylinder	Steam Lap	Exhaust Lap	Lead
8 to 12	3/16	1/16	1/32
14 to 20	1/4	1/8	1/32
22 to 30	5/16	3/16	3/64
32 to 36	3/8	1/4	1/16

3. Adjusting the Dash Pot Rods.—This is a very important adjustment. If the rods be too short, the steam valves will not open, *if too long, the rods will be bent, or the bonnets broken, or both.* To make this adjustment, the wrist plate is turned by hand to its extreme position, that is, until B, or C, of fig. 957 registers with D.

When the plunger is down as far as it will go, as in fig. 962, the dash pot rod D, should be adjusted so that the hook on shoulder H, will safely clear the steel die S, on the steam arm, leaving a margin M, between the hook die E, and steam arm die S.

4. Setting the Eccentric.—The paper which was inserted between the wrist plate and its washer may now be removed so that the wrist plate can turn. After doing this, the engine

*NOTE.—The reason for the increase in lap is to preserve the same angular advance of the eccentric for engines of different sizes. In order to do this, the lap must be increased in proportion to the travel.

is placed on the dead center and the eccentric located "by eye." The lead is measured and the engine turned over to the other center. If the length of the eccentric rod be correct, both leads will be the same; if not, the lead must be equalized by adjusting the eccentric rod to the correct length. The equalized lead will probably be too great or too little since the eccentric was set by eye; it remains then to change the position of the eccentric until the valves show the desired lead as given in the table.

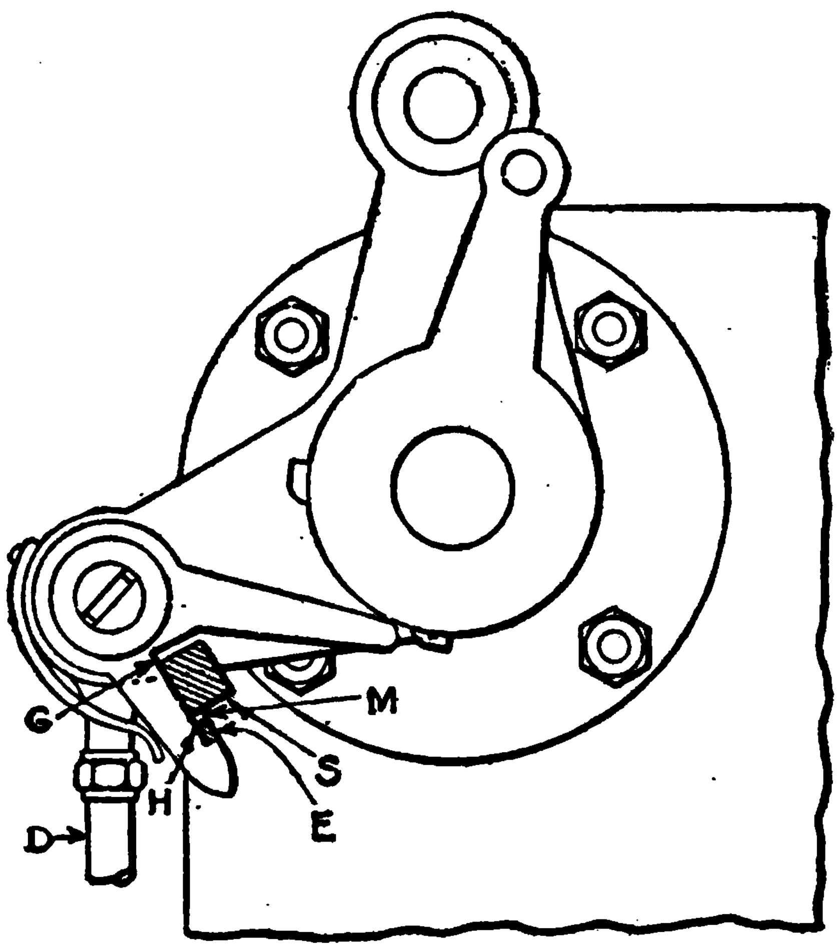


Fig. 962.—Detail of valve gear illustrating adjustment of dash pot rod.

The foregoing, of course, applies to engines having the eccentric fastened with set screws. In some cases, especially on large engines, when the eccentric is keyed to the shaft the position of the eccentric is fixed, and the steam lap as given in the table is subject to correction.

After the wrist plate, valves and rocker have been squared, and the lead equalized, if the correct lead be not obtained, the engine is placed on each dead center, and the length of the steam rods (K, K, fig. 961) adjusted until the valves show the proper lead as given in the table.

If for any reason it be desired to change the position of the eccentric as for instance, to increase the range of cut off, it may be done by fitting an offset key.

The setting of a Corliss eccentric is governed by the same principles that

FIGS. 963 and 964.—Rice and Sargent Corliss inlet valve gear in extreme positions. Fig. 963 shows the gear for the front end of the cylinder in its extreme left and opening position. The latch A, on the valve stem lever B, is in the position of engagement with the toe C, on the rocker D. The pin E, connects through the intermediate rockers and rods with the steam eccentric on the engine shaft, and the pin F connects to a similar inlet gear at the back end of the cylinder. As the rocker D, moves to the right, the toe C, engages the latch A, moving the inlet valve to open, and raising the dash pot plunger which is connected to the pin P. Cut off is accomplished by the toe C, turning downward on its pivot spindle H, to release the latch A. The spindle H, has a cam lever I rigidly attached in the rear, which in turn is carried between two hardened steel rolls, J, J. These rolls turn on pins in the cut off collar K, which latter turns freely on the valve stem journal. The arm L, above, forming part of the same casting as the cut off lever K, is connected to the governor by the rod M. This rod is held firmly by the governor and does not move unless there is a change in the speed of the engine. The rod N, connects to the valve gear at the head end of the cylinder. The latch A, is released at some point in the opening movement of the rocker D, toward the right. This is accomplished when the rise O of the cam lever passes between the cam rolls J, J. It is obvious that the length of cut off depends upon the position of the cut off collar K, as controlled by the governor. The further to the left the collar K, the earlier the cut off. Fig. 964 shows the rocker D at the extreme right of its motion. Release has taken place and the valve is about to be closed by the pull of the dashpot. The valve then closes promptly and the lever B turns to the position shown in fig. 963. The cut off collar K is here shown in the position, giving nearly the latest cut off, which is about three-quarter stroke of the piston. On the return movement of the rocker D, the cam rolls, J, J, raise the cam lever I, and the toe C, to the engaging position. At the latter part of the movement of the rocker D, to the left, as the toe C, passes under the latch A, the latter is raised by the toe sufficiently to clear the same, and the latch then drops by gravity in front of the toe to the engaging position.

apply to the slide valve eccentric; the operations differ only in minor details. For instance, it is not necessary to place the Corliss eccentric "by eye" a little ahead of its correct position when equalizing the lead as both positive and negative lead may be easily measured from the reference marks. Instead of wedges for measuring the lead, this is conveniently done by means of dividers.

5. Adjusting the Governor Connections.—There are two *governor cam rods* H, H, by which the controlling action of the governor is transmitted to the knock off levers, as shown in fig. 961. By lengthening or shortening these rods, the point of cut off may be adjusted. In doing this, the governor sleeve is raised by means of the safety stop.

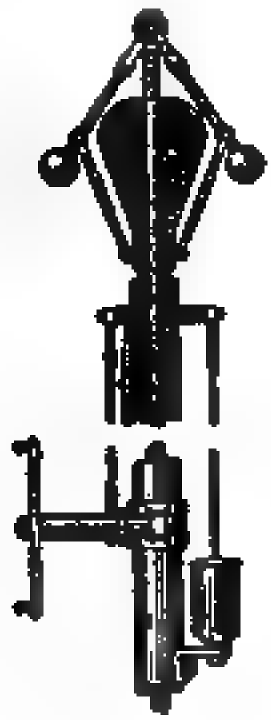
This prevents the governor reaching its lowest position, and brings it in the lowest position in which the hook should engage the steam arm.

With the governor in this position the carrier rod is unhooked and the wrist plate turned by hand to one of its extreme positions. The corresponding steam valve will now be wide open, and in this position the governor cam rod H, (fig. 961) is to be adjusted so as to bring the knock off lever G, in the proper position to release the steam arm M, thus allowing the valve to close.

The wrist plate is now turned to the other extreme position, and a similar adjustment made at that end of the cylinder.

To check the correctness of the cut off adjustments, the governor should be raised to an intermediate position and blocked. The carrier rod is then connected to the hook pin, and the engine turned over slowly in the direction in which it is to run, noting the positions of the crosshead at which each cut off takes place. If equal cut off be obtained for each stroke, no further adjustments are necessary, if not, the length of the cam rods should be adjusted until the points of cut off are at equal distances from the beginning of the stroke. The valve bonnets may now be replaced, and the block removed from the governor which completes the setting of the valves.

Setting Corliss Valves, Double Eccentrics.—The work of setting the valves of a Corliss engine having two eccentrics,



FIGS. 965 to 972.—Murray governors. Figs. 965 and 966 show outside and sectional view of plain governor. Figs. 967 and 968, variable speed governors. These were designed especially for Corliss engines for variable speed service, the speed adjustment being made by hand without stopping the engine. Figs. 969 and 970, special attachment, for hand adjustment of speed. The weight is moved in or out to increase or decrease speed. Speed range fig. 970 greater than that of fig. 971. Figs. 971 and 972. Cut off adjustment for compound engine. The conditions of service under which compound engines are worked are frequently such that make it very desirable that the point of cut off in the low pressure cylinder may be changed by hand to points earlier or later than that taking place in the high-pressure cylinder, while the engine is in operation. Figs. 971 and 972 show such device attached to tandem compound engine governor. By means of a hand wheel the position of the bell crank that carries the governor knock off rods of the low pressure cylinder is changed to any desired point, thereby changing the point of cut off in that cylinder and raising or lowering the receiving pressure correspondingly. This device may be thrown in or out at will. Fig. 972 shows the hand wheel disconnected and the bell [illegible] position where the point of cut off is controlled i[illegible] the governor. When this device is in use the governor has control of the cut off in the high pressure cylinder only, that of the low pressure remaining constant where set.

is not so complicated as supposed by some. It is rendered easy by keeping in mind the principles involved.

The reason for two eccentrics, as before explained, is to obtain a long range cut off without distorting the action of the exhaust valves.

If all four valves were operated by one eccentric, cut off could not take place later than one-half stroke without causing release and compression to occur too late. This is due to the negative angular advance which must be given the eccentric to secure late cut off.

With two eccentrics, then, *negative* advance may be given the steam eccentric to get a late cut off, and *positive* advance to the exhaust eccentric to secure early release and compression.

The arrangement of the steam rods of a single eccentric gear is such as to give a slow initial* movement to the valve while the port is still covered with positive lap; this causes the quick motion period of the valve movement to occur while the valve is opening the port.

The steam rods of a double eccentric gear are arranged to give a quick initial valve movement, because negative lap is used here, which causes the valve to open the port at an earlier point in its travel.

It follows then, that a valve gear designed to be operated by a single eccentric cannot very well be made to cut off much later than half stroke, even if a separate eccentric be added, because the slow initial movement of the single eccentric gear cannot be corrected without a re-arrangement of the steam rods.

In setting the valves of a double eccentric gear, the operations of squaring the valves, equalizing the travel, etc., is practically the same as with the single eccentric engine. The various steps are as follows:

1. *Squaring the wrist plates and rockers;*
2. *Squaring the valves;*
3. *Adjusting both eccentric rods, equalizing the travel;*
4. *Adjusting the dash pot rods;*
5. *Setting the exhaust eccentric;*
6. *Setting the steam eccentric;*
7. *Making the governor adjustments;*
8. *Final adjustments with indicator.*

---

*NOTE.—The initial movement of the valve means the beginning of its movement starting from the extreme position.

1. Squaring the Wrist Plates and Rockers.—These are squared in the same way as with the single eccentric gear. That is, the wrist plates are placed in the neutral position as indicated by the reference marks and clamped; the setting is then verified with the plumb bob. Similarly, after unhooking the carrier rods, the rockers are placed in a vertical position and the carrier rods adjusted to the proper lengths, using plumb bobs as in fig. 959.

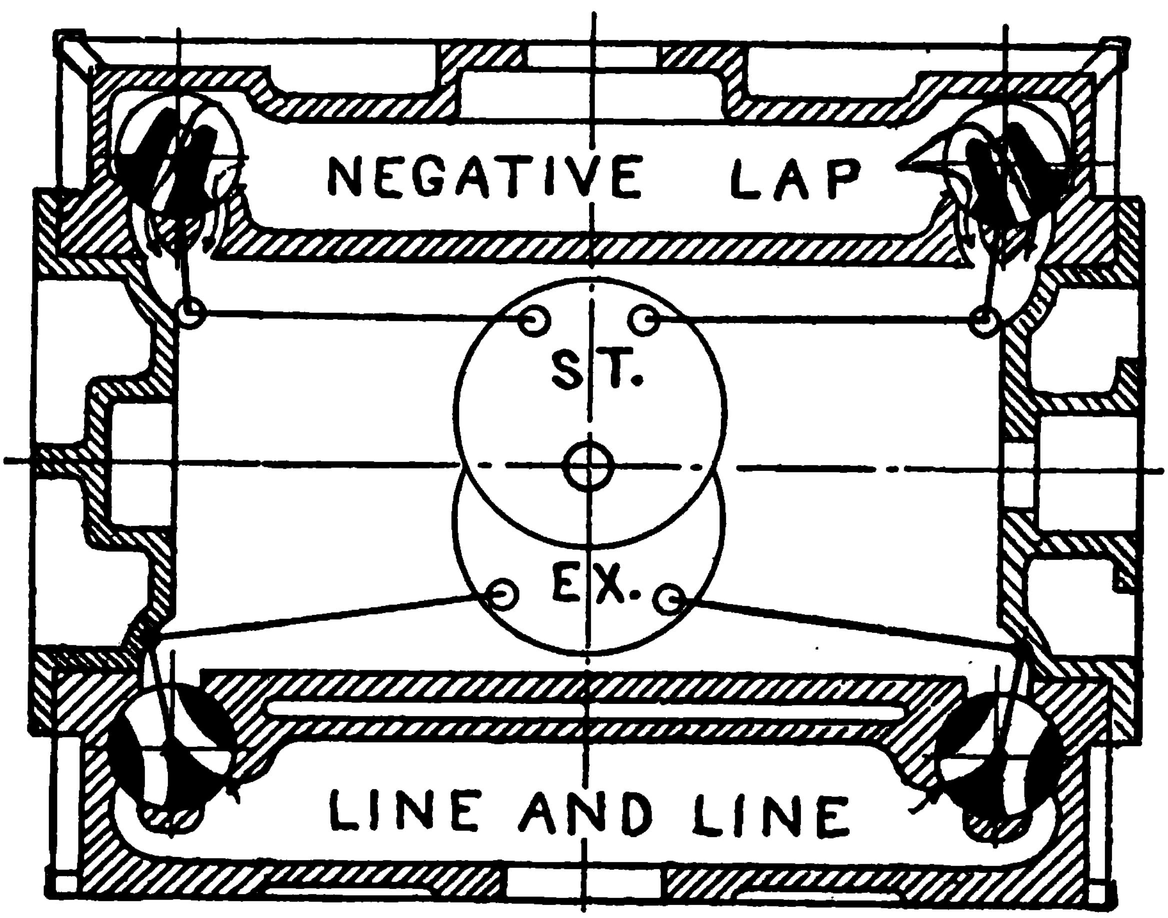


FIG. 973.—***Squaring the valves with double eccentrics.*** The steam valves are set with negative lap which is usually a little less than half the port opening. The usual setting of the exhaust valves is line and line as shown.

In squaring the rockers, the eccentrics should be unloosened on the shaft and the rockers moved by turning the eccentrics rather than the engine.

2. Squaring the Valves.—These are squared by adjusting the steam and exhaust rods the same as with the single eccentric gear. However, with two eccentrics, the steam valves are set with *negative lap*, which, in amount, is usually a little less than half the port opening as shown in fig. 973.

The object of this negative lap is to so locate the position of the eccentric that it will give a quick movement to the valve in opening the port.

The exhaust valves are set in *line and line* position as illustrated in the figure.

3. Adjusting the Eccentric Rods.—The travel of the valves should be equalized, by adjusting the lengths of the eccentric rods. After unclamping the wrist plates, each eccentric is turned on the shaft so as to bring the wrist plates in the extreme positions, noting if the reference marks register at these points. If not, the eccentric rods are to be adjusted until the marks come opposite each other.

In case the eccentrics are keyed, or not easily turned on the shaft, the engine may be turned over instead, in equalizing the travel. *If this be done, great care should be exercised to see that the dash pot rods are not too long, otherwise the valve gear may be injured as previously explained.*

4. Adjusting the Dash Pot Rods.—The length of these rods is adjusted in the same way as with the single eccentric gear, hence no additional instructions are necessary. It is well to repeat, however, that this is an important adjustment and should be carefully made. *If the rods be too short, the steam valves will not open; if too long, the rods will be bent, or the bonnets broken, or both.*

5. Setting the Exhaust Eccentric.—This is usually set first as it is next to the shaft; the other eccentric then may be turned out of the way if necessary in tightening the set screws.

To locate the position of the exhaust eccentric, the engine is turned *in the direction it is to run* until the piston is brought to the point where compression should begin.

The distance of this point from the end of the stroke, may be taken at 5 per cent of the stroke. This is easily measured by scribing reference marks on the cross head and guide.

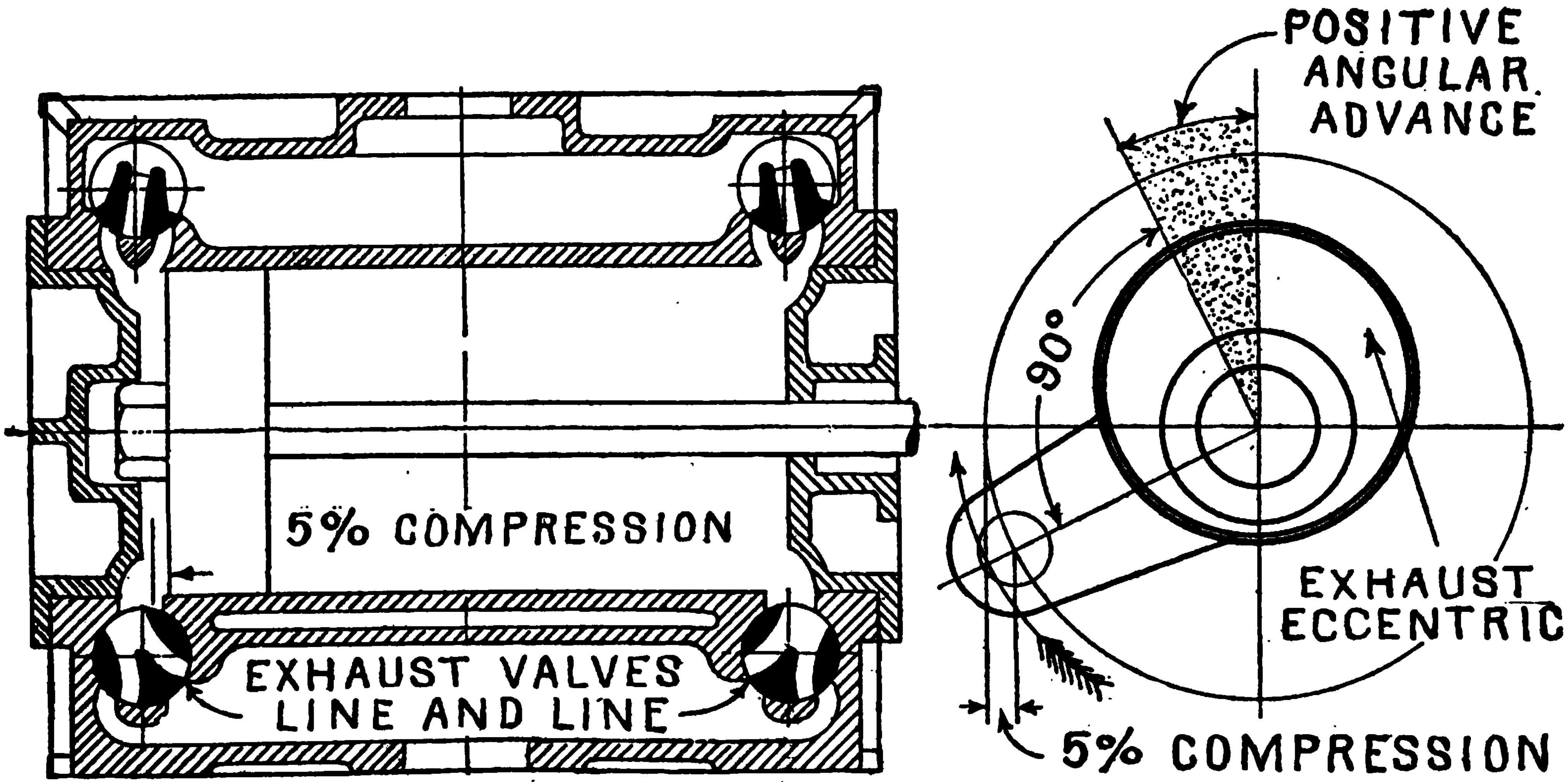
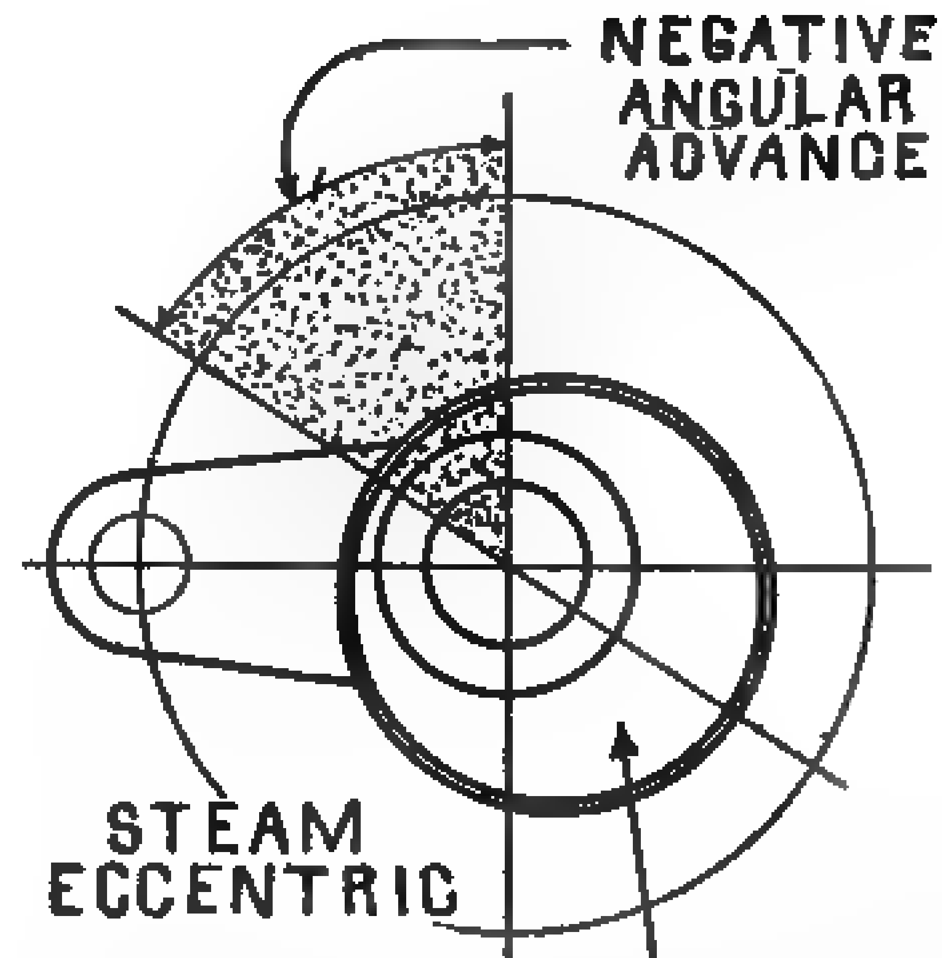


FIG. 974.—***Setting the exhaust eccentric.*** With the piston at the desired point of compression, the exhaust eccentric is advanced until the exhaust valves come in line and line position. The positive angular advance which is given the eccentric is indicated by the shaded angle.

The eccentric is now turned in the direction the engine is to run, until the exhaust valves are line and line as shown in fig. 974, and then fastened in position on the shaft.

The diagram to the right of the figure shows the relative positions of the crank pin and exhaust eccentric.*

*NOTE—For clearness the steam eccentric is omitted in the diagram.



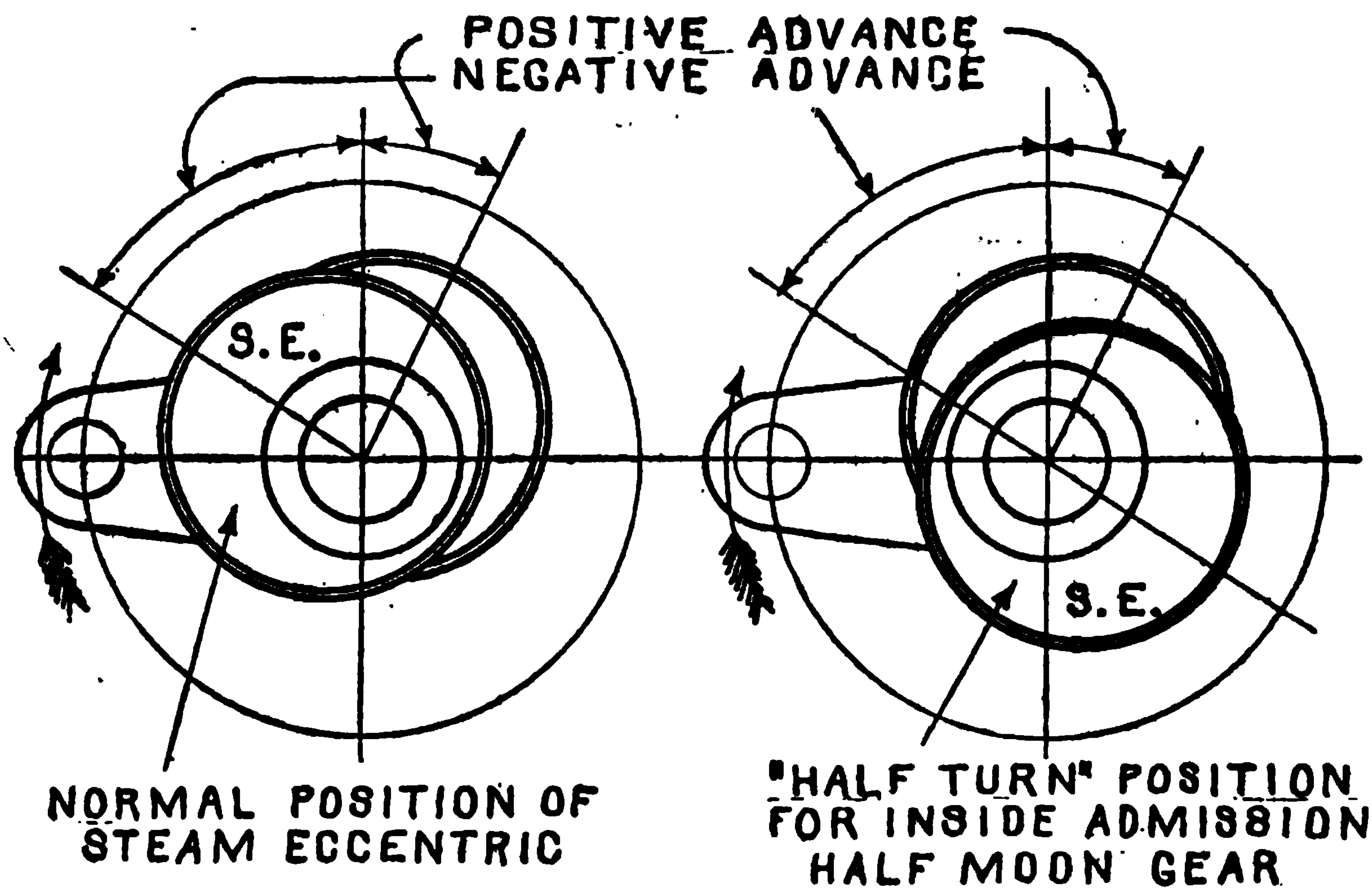
"HALF TURN" POSITION

dead center and the eccentric advanced until the valve
ar advance will be negative as indicated by the shaded
rn" position, that is, 180 degrees from the position of

angular advance.

6. **Setting the Steam Eccentric.**—The engine is now placed on the dead center, and the steam eccentric turned in the direction *in which the engine is to run* until the valve has the proper lead, as shown in fig. 975; for a double ported valve this will be less than required for the single ported type. If the eccentric rod adjustment previously made be correct; the lead for the other end of the cylinder will be the same, otherwise a further adjustment of the eccentric rod is necessary.

It should be noted that the steam eccentric will be either in its normal position of negative angular advance, or at a half turn (180° degrees) from this position, depending upon the type of the valve gear. Thus, with the half moon gear and inside admission as shown in fig. 977, the motion of the valves must be reversed, hence the steam eccentric must be given a half turn to the right as shown in the figure. If these valves had outside admission, the steam eccentric would be in its normal position as shown in fig. 976.



FIGS. 976 and 977.—Illustrating ***normal,*** and ***half turn*** positions of the steam eccentric. The eccentric is placed in its normal position when its center moves in unison with the hook pin on the wrist plate, and at half turn position when they move in opposite directions. For example, the eccentric is set in its normal position for the crab claw and oval arm gears, and at half turn position for an inside admission, half moon gear.

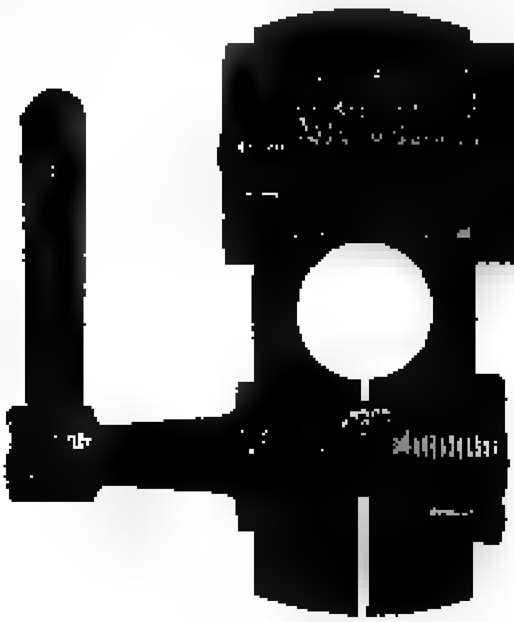
7. Making the Governor Adjustments.—The method of adjusting the governor with the double eccentric gear is the same as with single eccentrics, as described on page 515.

8. **Final Adjustments with Indicator.**—It is desirable after setting the valves that an indicator be applied to the engine when it is in operation, to verify the valve setting, and make more accurate adjustments than would otherwise be possible. In using the indicator the following directions for the final adjustments should be noted:

what worn.

If with the average load on the engine, the indicator cards from the two ends of the cylinder be unequal, showing that one end is doing more work than the other, the governor rods should be adjusted to cut off a little earlier at the end with the larger load, and a little later at the other end.

If the toe of the diagram turn up, showing that pre-release is too late, the exhaust rods, with single eccentric gear, may be shortened a little, not forgetting that this is at the expense of reducing the compression. Lengthening the rods will increase compression, but make release later.



Figs. 979 and 980.—Filer and Stowell hook and device for disconnecting the valve gear from the eccentric motion.

With the double eccentric gear, a later release should be corrected by giving the exhaust eccentric more positive angular advance.

If the card appear a little late all around, the eccentric (or eccentrics) should be set a trifle forward.

Any small changes in the admission lines that may be desired without affecting the rest of the diagram are made by adjusting the steam rods. Shortening the steam rods will give earlier admission, while lengthening them will, of course, produce the opposite result. Figs. 838 to 848 (page 461) in the chapter on Valve Setting show effect of errors in valve setting as recorded by indicator diagrams.

CHAPTER 15

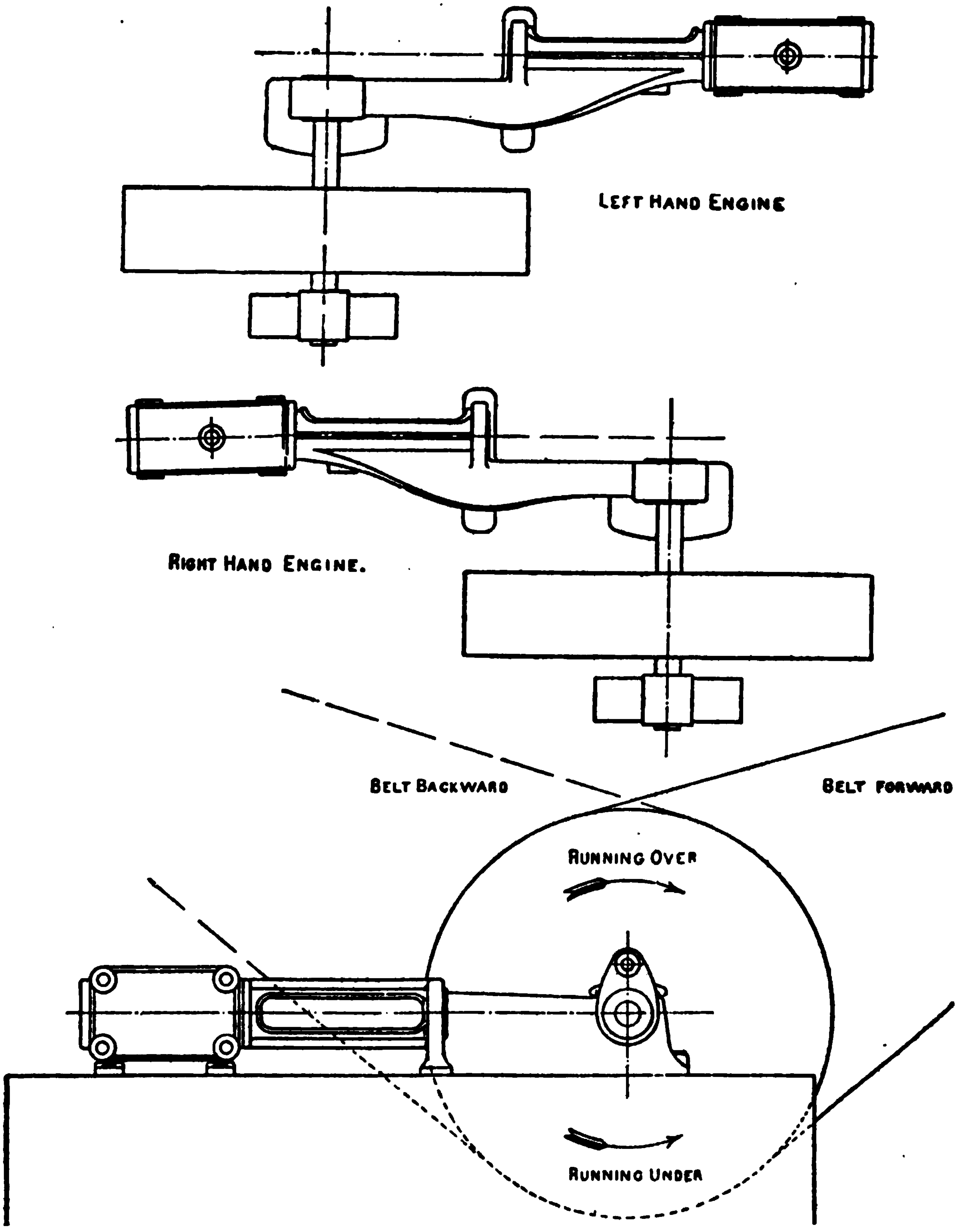
HOW TO RUN A CORLISS ENGINE

Before Starting.—The engineer should first examine the engine parts, and satisfy himself that every thing is in working order before attempting to start. The cylinder lubricator, and oil cups should be filled and set for the proper feed. The cylinder should be lubricated with the best grade of cylinder oil, the choice depending on the temperature of the steam. For high pressure, or superheated steam, care should be taken to select an oil having a sufficiently high flash point.

The surfaces of the cylinder and valves will be improved by the use of flaked graphite prepared for the purpose; this can be mixed with the cylinder oil and injected with a hand pump, or fed clear in a cup especially designed for it. Graphite, like lard oil, is excellent for hot bearings, besides being useful in other ways; it should be included in the list of engine room supplies.

In packing the piston rod, and valve stem stuffing boxes, care must be taken to put in the packing evenly, so that steam may be prevented escaping without the necessity of screwing down the glands so tightly as to interfere with the free movement of the rods.

The packing should be changed as often as may be necessary to prevent it becoming hard and gritty; keeping it too long in use may be the means of grooving or creasing the rods, thereby occasioning leakage and consequent trouble.



FIGS. 981 to 983.—Right and left hand Corliss engines, and diagram illustrating the term *running over* and *running under*.

Before starting the engine, the drain cock on the steam pipe should be opened, and any water that may have accumulated, blown out. After placing the governor safety stop in starting position, the carrier rod is unhooked, and the *starting bar* inserted in the wrist plate. With the throttle valve slightly opened the starting bar is moved back and forth a few times allowing a little steam to pass into the cylinder until it becomes thoroughly heated and freed from water.

Starting.—The engine should be slightly off the center and the proper valve opened for admission by means of the starting bar. The throttle is now opened sufficiently to allow the engine to gain enough momentum to pass the other center. Just before reaching the end of the stroke the wrist plate is turned to the other extreme position with the starting bar so as to open the opposite admission valve for the return stroke. The valve gear is thus worked by hand with the starting bar (the carrier rod being still unhooked) and the engine run slowly in this way a few revolutions.

The reason for working the valve gear by hand in starting is that the steam valve may be fully opened for admission at the beginning of the stroke, thus giving free entrance to the steam; this is especially desirable in starting with a load.

The carrier rod is now hooked to the wrist plate stud and the engine gradually brought up to full speed.

In some cases where there is very little load at starting, hand control is not necessary after the preliminary warming up process.

It now remains to put the safety stop in running position, and it should be noted that *failure to do this may result in a wrecked engine, and loss of life.*

Before starting an engine for the first time, the valve gear should be tested by working it with the starting bar to see if

FIG. 40.

FIGS. 984 and 985.—Wheeler counter current jet condenser; fig. 984, top exhaust inlet counter current type with submerged pump; fig. 985 vertical low level type with side exhaust inlet and submerged pump.

it will move through the extent of its travel without bind or interference anywhere, *special attention being given to the dash pot rods to see that they are not too long.*

Starting: Jet Condensing.—In starting a jet condensing engine, care should be taken to follow certain rules, to prevent water entering the cylinder with its attendant dangers. There are two methods of procedure, depending on whether the engine has:

1. An independent air pump; or,
2. A direct driven air pump.

With an independent air pump, the injection valve is opened slightly and the air pump started to its normal speed. When the vacuum is established as indicated by the gauge, the engine is warmed up in the usual manner, and started with the hand bar. As the engine is being brought to speed after hooking on the carrier rod, the injection valve is regulated so that the supply of cooling water will be sufficient to condense the steam, otherwise the vacuum will fall.

The amount of cooling water and speed of the air pump must be regulated according to the degree of vacuum required, *being careful that the air pump is running fast enough to take care of all the water admitted.* It is therefore important to know when the proper supply has been reached. The engineer is guided in this by the vacuum gauge. As the injection valve is being opened, the vacuum will increase up to a certain point after which any additional opening of the valve will not increase the vacuum. This indicates that the pump is receiving all the water it can handle, and any excess would tend to flood the condenser. *The condenser should* **not** *be operated with the injection valve opened to this extent*, but should be closed a half turn or so, or until the vacuum begins to fall to guard against exceeding the capacity of the pump. If a higher vacuum be desired, the speed of the pump must be increased to take care of the larger amount of cooling water required.

A steam by pass should be fitted to the exhaust pipe at the engine, especially when the supply of cooling water is at a lower level than the condenser in order to facilitate the formation of a vacuum by blowing out the air, and priming the condenser with steam.

In starting an engine having a direct connected air pump, the cylinder is first warmed, and the engine set in motion before opening the injection valve. This allows the condenser to fill with steam which displaces the air. As soon as the engine is in motion the injection valve is slightly opened, the full supply of cooling water being not admitted until the normal speed has been reached.

The reason for this is on account of the air pump being direct connected, its speed will vary with that of the engine, and while the engine is

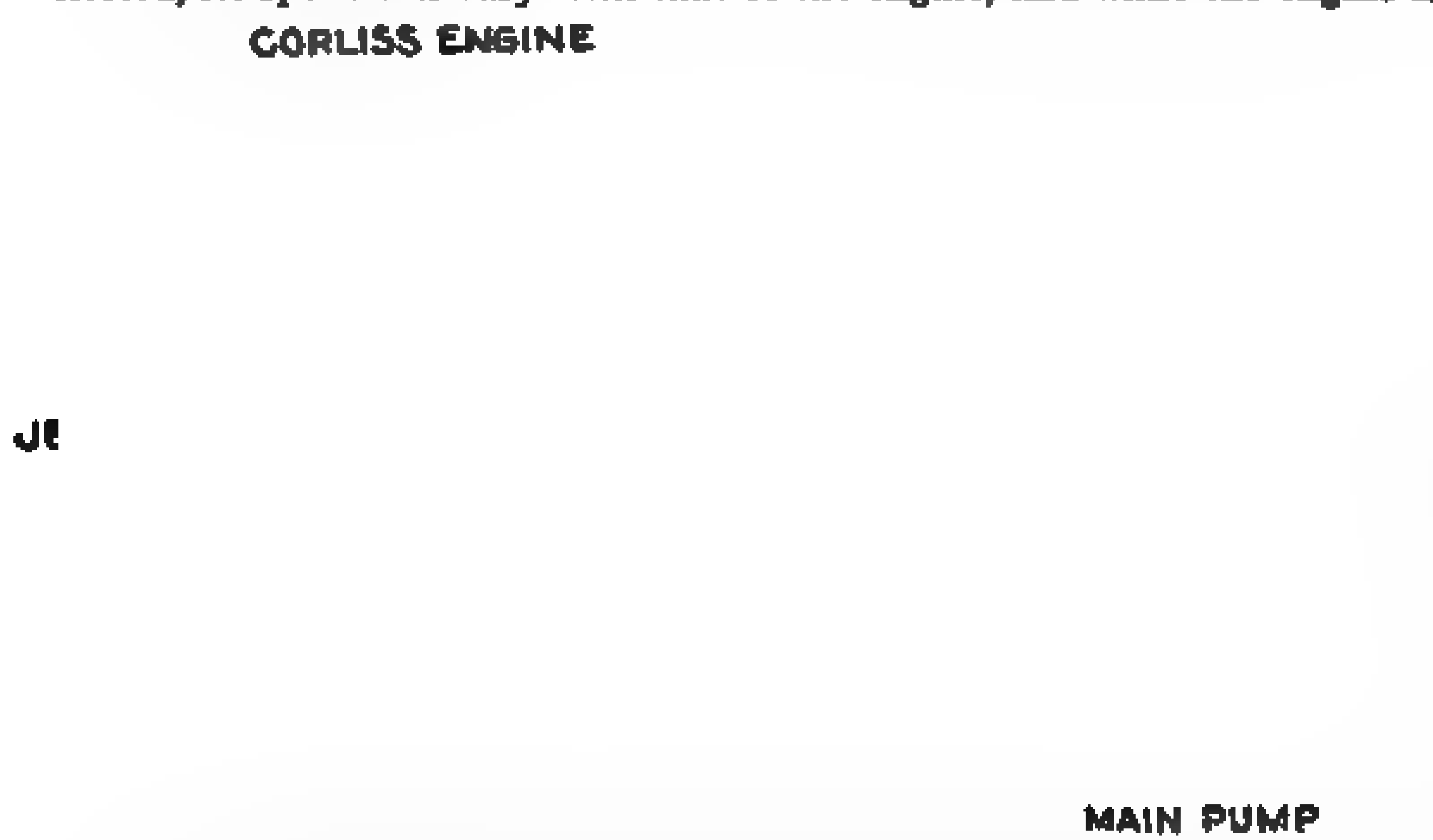


Fig. 986.—Corliss engine with Dunham direct connected pump. As piped, a water heater is placed between the engine and jet condenser, the latter being attached to the pump.

running slowly, the pump displacement would not be sufficient for the full supply of cooling water. The condenser under these conditions might flood and the water back up into the cylinder.

In the chapter on "Engine Management" additional instructions will be found on starting with various types of condensers.

While Running.—When an engine is in operation, the careful engineer will give the proper attention to lubrication, to see

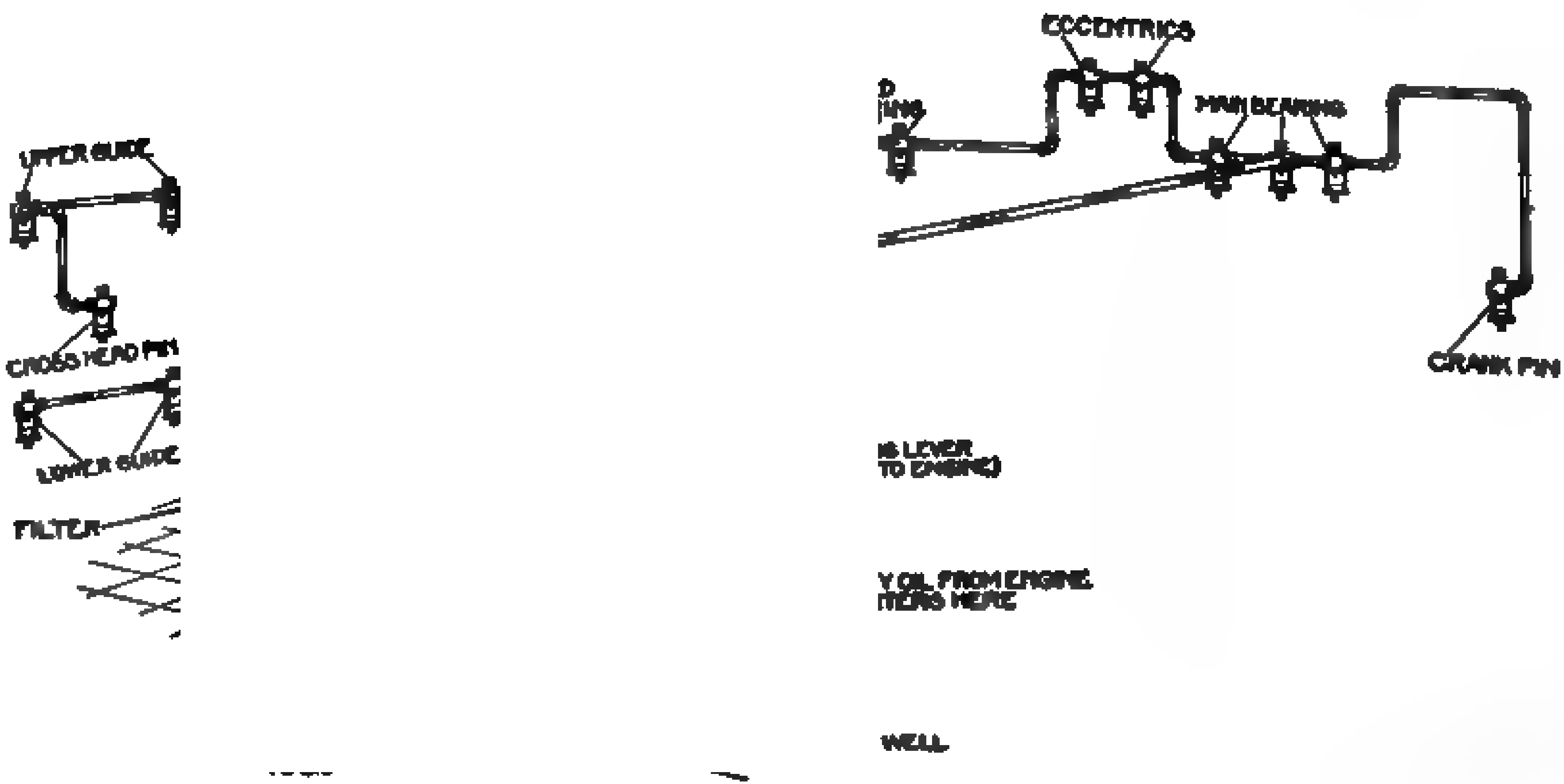


FIG. 987.—Richardson individual oiling and filtering system. It comprises a filter with overflow stand attached, double ended pump and cast iron receiving well with necessary piping and sight feed oilers.

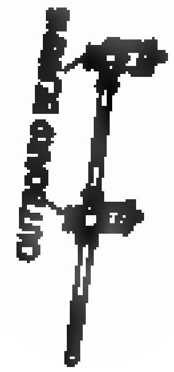
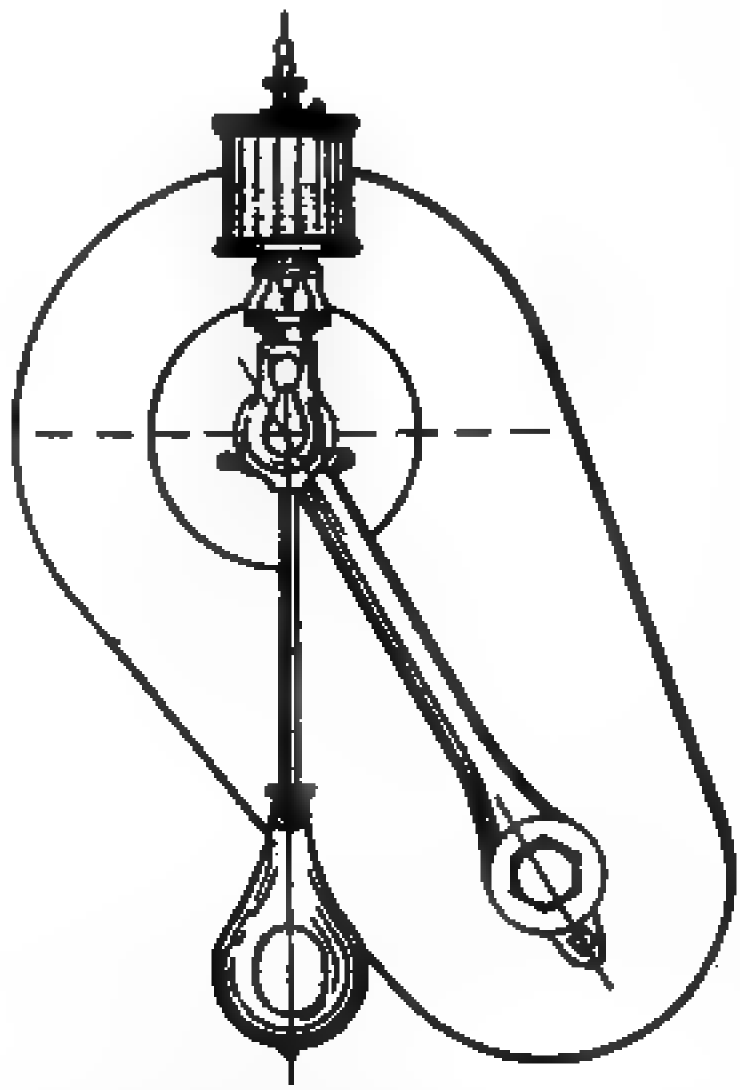


FIG. 988.—Phoenix individual oiling and filtering diagram, showing apparatus and material necessary for equipping a simple engine having 10 points of lubrication.

that none of the bearings are heating, and keep an alert ear for any unusual noise.

A careful study should be made of the oil requirements so that each bearing will receive the proper quantity *and no more.* In places not provided with automatic oil cups, it should be remembered that oil applied frequently, and in small amounts is more efficient than spasmodic flooding.



FIGS. 989 and 990.—Side and front views of Nugent crank pin oiler. The tube or arm through which the oil flows by centrifugal force to the crank pin is bolted solid to the end of the crank pin and is half the stroke of the engine in length, so that one end is always concentric with the axis of rotation. On this latter end is journaled an oil cup holder weighted with a pendulum bob which keeps the oil cup stationary and in an upright position. The cup holder is held in place by a cotter pin extending into a groove cut into the journal. The arm consists of the tube, bowl, journal and end, which bolts to the crank pin. The bolt is dulled, as shown, connecting the oil conduit in the arm to that dulled in the crank pin.

To a person accustomed to the regular sounds of the engine room, the ear is quick to detect any unusual noise as a knock, or pound. Sometimes assistance in the detection of such may be had by the use of a convenient piece of metal, as a piece of pipe, or spanner, one end being placed to the ear, and the other against that part of the engine where the trouble is supposed to be located.

Knocking is usually caused by loose adjustment, a loosened nut or bolt; in some cases, caused by parts of the engine being out of alignment, or a shoulder worn at the cylinder end by the piston.

Heating may be due to insufficient lubrication, poor adjustment, or bad condition of the bearings, and is detected by the sense of feeling or smell.

Fig. 991.—Main bearing of the Rice and Sargent Corliss engine. It is of the four point type, having bottom, side, and top shells, and both side shells are adjustable by means of guided wedges which, as shown, drop to release the shaft, thus avoiding a possible injurious pinching. The bottom shell rests in a semi-circular seat, making it possible to remove the bottom shell by slightly raising the shaft. Large sizes are water cooled. Under the bottom shell and in the metal of the bed itself, except in the largest sizes, is a large oil reservoir from which oil is fed continuously to the shaft by means of the chain oilers, as shown. A drip is provided to draw this oil off when desired. Proper scoring and grooves are provided to secure proper distribution of the oil to the journal. The shells are lined with Babbitt metal dovetailed into place. For inspecting the shaft, the top shell and cap are provided with hard holes fitted with cast iron covers.

Most bearings may be felt by the hand, being cautious to avoid accident. If a bearing heat beyond a moderate degree, the oil will give off a burnt odor, and in severer cases, will smoke freely, giving visible evidence of trouble.

Small leaks at stuffing boxes, joints, etc., should not be neglected, as every leak means a loss of fuel. In adjusting the valve

fire test and medium body or viscosity should be used, while for low pressure, an oil of low fire test and heavy body is required.

If an engineer running a compound engine, have only the high fire test oil, he can sometimes make it right for the low pressure cylinder by the addition of ordinary lubricating oil. The mixture is sometimes improved by the addition of a table-spoon full of *clean* tallow to the quart. Tallow, however, should be used with caution, and not as a regular lubricant because it contains acids which attack the metal of the cylinder.

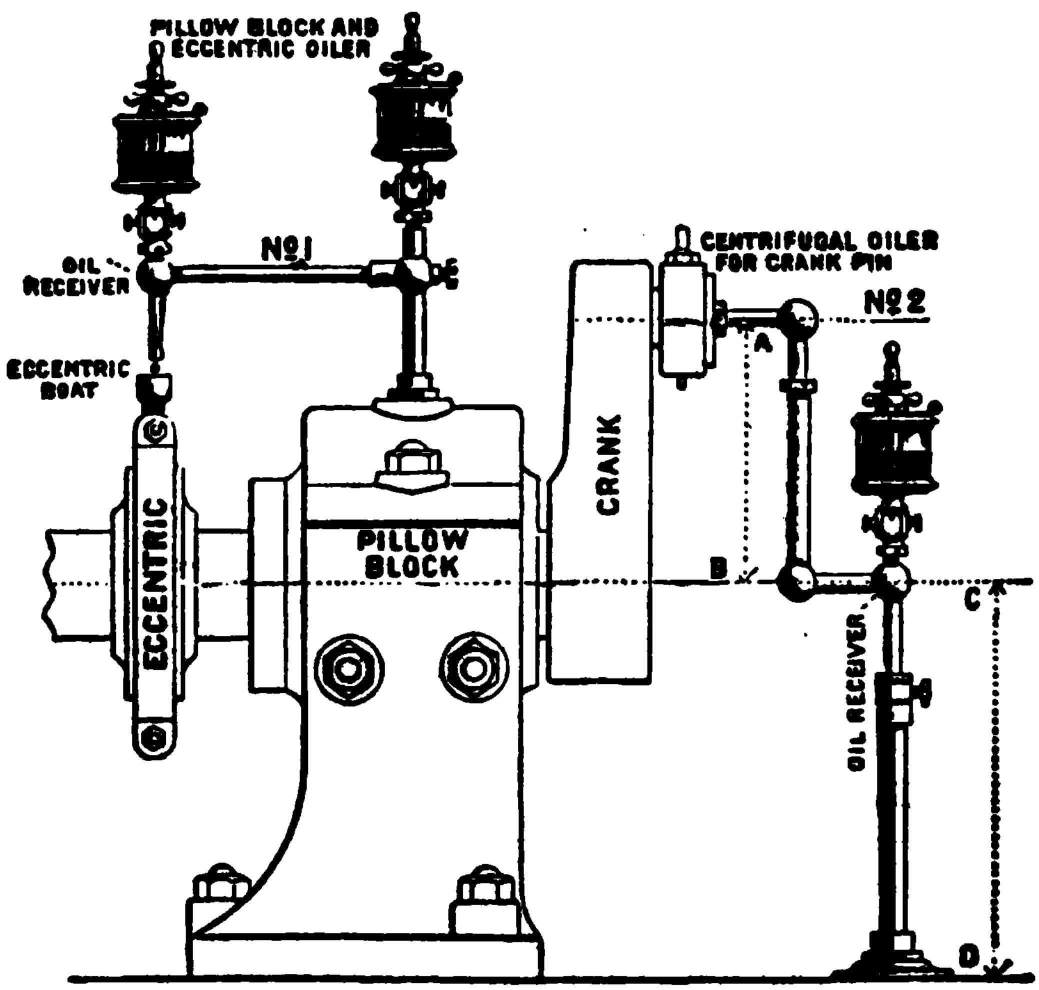


FIG. 994.—Powell oiling devices for crank pin, pillow block and eccentric.

To test for animal oil, a two ounce bottle is filled half full of water, and a stick of caustic soda or potash, or a little strong ammonia added. A sample of the oil is poured in the bottle and the contents well shaken. If animal oil be present, it will separate, and leave the mineral oil intact, except when compounded in special ways with Neatsfoot oil when the mixture will thicken. This test may be used to indicate if lard, sperm, or any animal oil is adulterated with cheaper mineral products.*

*NOTE.—Considerable additional information on oils and how to test and apply them will be found in the chapters on Lubricants, and Lubrication.

indicates that the minimum oil supply has been reached.

This method applies to bearings with no provision to catch the waste oil. Where a receptacle is provided for drain, no such refinement is necessary.

The main bearing of Corliss engines is usually fitted with an oil well and chain distributer as shown in figs. 991 to 993. The attention here necessary is to maintain the oil in the well at the proper level, and to frequently filter same.

The base plate of the outboard bearing is usually provided with a surrounding rim forming a receptacle for waste oil. This plate should be tapped and fitted with a small pipe, to drain off the oil as it accumulates; it should then be passed through a filter before using again in the oil cup.

FIG. 998.—Screw feed marine grease cup. This cup is designed for the main bearings of marine engines, but will also be found suitable for other purposes where a screw feed is desired, such as for forcing grease some distance to the parts to be lubricated.

The engineer should not neglect to give attention to the numerous bearings of the valve gear and keep them properly lubricated.

Hot Bearings.—When the ordinary remedies such as water, lard oil, and easing up the brasses fail, an application of white lead mixed with machine oil to the right consistency will prove helpful. A small can of white lead should be kept on hand for this purpose.

Oil Required for Cylinder.—It would be safe to state that the average stationary engineer, having charge of a small or

medium size Corliss engine will use from 4 to 10 drops of cylinder oil per minute or 4 to 10 times too much. Two examples here given, and known to the author, demonstrated this:

A 16×42 Reynolds Corliss engine has been operated for a considerable period with one drop of oil per minute.

A 10 and 16×12 Thropp fore and aft compound marine engine having slide valves was run with one drop every five minutes without ill effect. The latter example shows how little oil is necessary for satisfactory engine operation.

FIG. 999.—Plain cylinder lubricator without sight feed.

FIG. 1,000.—Graphite sight feed lubricator. *To operate*, close steam valve and open drain plug to allow steam to escape from cup; then close regulating valve, remove filling plug and fill cup with graphite. After replacing filling plug, close drain plug, open steam valve (wide) and regulate the feed of graphite by regulating valve. The sight feed glass can be cleaned by opening drain plug. If necessary to replace the sight feed glass, take cup apart by means of lock nut, and slide the new glass down through the opening.

A horizontal cylinder, of course, will require more oil than a vertical one, and a releasing gear, more oil than the positive cut off type.

The engineer, in determining the rate of feed for any particular engine must, of course, be guided by the behavior of the engine and his judgment in the matter, keeping in mind that where a surface condenser is used and the condensate returned to the boiler, *as* **little** *oil as possible should be used.*

NOTE.—The Vacuum Oil Co., claim for their "600 W" Cylinder Oil, one drop per minute to one drop in two minutes, as sufficient for a 20 and 36×48 compound Corliss engine running 83 R. P. M.

Contrary to the popular belief, the effect of wet steam in a cylinder does not act as a lubricant. This has been demonstrated by the performance of a Corliss engine working with saturated and with wet steam. Although the valve gear operated smoothly when the steam was dry, if the boiler primed, and water

FIGS. 1,001 and 1,002.—Detroit sight feed lubricator. The parts are A1, body of oil reservoir; A2, condenser; A3, filler plug; A4, water feed valve stem; A5, plug for inserting sight feed glass; A6, sight feed glass drain stem; A7, sight feed regulating valve stem; A8, drain valve; A9, globe valve in support arm; A10, plug for inserting gauge glass; H, sight feed glass; J, gauge glass; K, connection to steam pipe.

was carried into the cylinder, the dash pots failed to entirely close the valves, thus showing the non-lubricating effect of the water when the valve was in an unbalanced position.

Knocks and Pounds.—The familiar click due to water being

reversals of thrust of the cross head near the ends of the stroke require

a refinement in the adjustment of the cross head gibs to avoid knocks at these points.

Vertical engines of the direct connected type usually have the fly wheel, and revolving member of the generator mounted on the shaft between the engine bearings; with this construction the total weight of the rotating parts generally becomes so heavy that the shaft is not lifted from its bottom bearing by the upward pressure on the piston. It is therefore desirable to allow an unusual amount of clearance between the shaft and the top bearing, as this permits a freer distribution of the lubricating oil. If, therefore, the pressure in the cylinder should from any cause be increased

FIG. 1,004.—Bates main bearing. It has adjustable boxes on each side of the shaft which are held in place by four wedges. The wedges can be raised or lowered by means of threaded bolts. A suitable locking device clamps the wedge bolts securely at any desirable point. By slightly raising the shaft, the bottom shell can be rotated around the shaft and removed.

to a point sufficient to lift the shaft, the result will be a heavy pound when it returns to the bottom bearing. The above occurs in case water accumulates in the bottom end of the cylinder.

Should the receiver pressure in a compound engine become excessive the shaft will lift; the same thing may also occur from excessive compression.

If an engine be running condensing, with the valves adjusted properly for that condition, then under non-condensing conditions the compression would be too great, which would tend to lift the shaft and cause pounding.

If the weight on the bearings be not in excess of the upward force on the piston, the bearing caps of vertical engines must, of course, be set up snugly. A good example of the latter case is the marine engine, which has no fly wheel, and the weight on the bearings is that of the shaft only.

The knock produced by an eccentric strap which is too loose usually has a slapping sound unless the speed be high, or lost motion great. The reason for this is that the bearing surfaces of the eccentric and strap are large in proportion to the forces transmitted. In adjusting an eccentric strap, the engineer should be careful not to get it too tight, for while the pressure per unit of area is small, the sliding velocity is usually high. This is especially true in direct connected engines, where the shaft is of large diameter.

Corliss Valve Knocks.—These may be located by flooding with oil. Since the valves are free to lift from their seats, a knock will result when, either on account of excessive compression, or the presence of water sufficient to fill the clearance space, the pressure in the cylinder becomes greater than the pressure in the steam chest. The valve will then lift from its seat as the piston approaches the end of the stroke, acting as a safety valve and return with a slam when the pressure drops.

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If a condensing engine, be run non-condensing, the early compression might produce a terminal back pressure greater than that in the steam chest, lifting the steam valves and causing a knock.

If the exhaust valves be free to lift, a non-condensing engine running without load may cut off so short that the steam will be expanded considerably below atmospheric pressure. The steam in the exhaust pipe, then, being of higher pressure, will lift the exhaust valves, before release rush into the cylinder, and cause the valves to rattle.

The end play of Corliss valves sometimes causes knocks. These valves are round and fit the chamber quite snugly for some distance at each end. One end is slotted for the stem, with the result that there is more waste space for steam than at the back end. Now, when the exhaust valve

B

FIG. 1,030.—Remedy for a knock caused by the end play of Corliss valves. By cutting in the valve small passages A, B, the accumulation of pressure between the valve ends and bonnets is prevented.

opens, the steam in this waste space expands and drives the valve against the back bonnet, sometimes causing a severe knock, even when the clearance is small. When the valve has closed and steam is admitted, the small space at the back end will accumulate pressure faster than the larger space at the stem end, and the valve will be driven back. The end knock of an exhaust valve can be distinctly felt by holding any object firmly against the back bonnet.

This defect may be corrected, as shown in fig. 1,030, by cutting passages for the steam in the ends of the valve, large enough so that the pressure will always be the same at the ends of the valves as in the cylinder.

Insufficient lubrication will cause valves to chatter on their seats, or too much spring in the various parts operating the valves, allowing them to

move by jumps, as it were. Sometimes part of this chattering is caused by the valve motion having loose joints, giving the valve a jerky motion. In some cases it will be found that while ample oil is being supplied it does not find its way between the valve and seat. Grooving the face of the valve or seat will often aid the distribution of the oil.

In large engines, excellent results have been obtained by grooving the valve seat and piping the oil into grooves under the valve, when no amount of oil sprayed into the steam would stop the grating of the valves. An inexperienced person might mistake this chattering of the valves for the piston "grunting for oil," but the former can readily be detected by the slight trembling of the valve connections.

Should the piston become loose on the rod, even a small amount, a heavy knock will result, which usually gets worse rapidly. The most com-

FIG. 1,031.—Cross section through cross head and bed of Rice and Sargent engine showing construction, rocker arms, lubrication of guides, wrist pin, etc.

mon practice is to have the piston held by a nut on the rod. This may not have been tightened sufficiently, allowing it to back off. It also happens that pistons are loosened by water in the cylinder.

Any striking of the piston or the studs against the cylinder head can be readily felt, by holding any object, as the end of a lead pencil, against the cylinder head. Whenever any knock occurs within the cylinder it

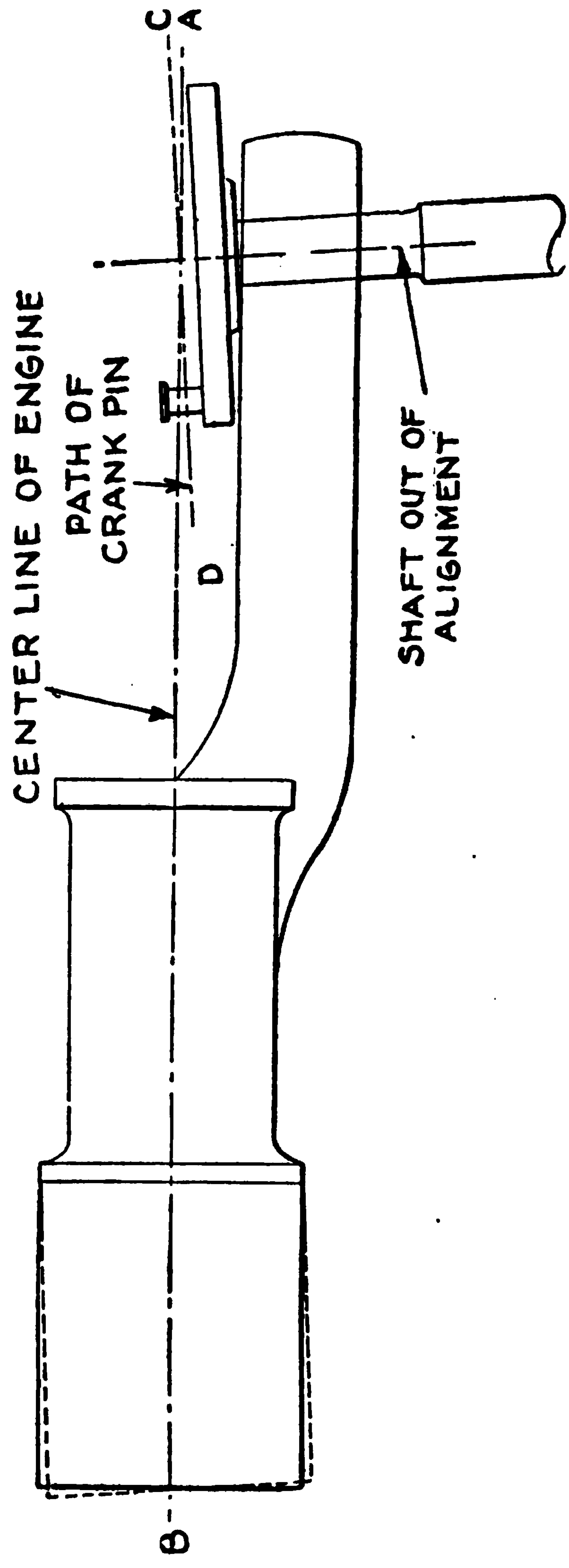


FIG. 1,032.—Bad alignment: shaft out of line with the cylinder axis causing a side slap or knock at the crank pin if the brasses have any side clearance.

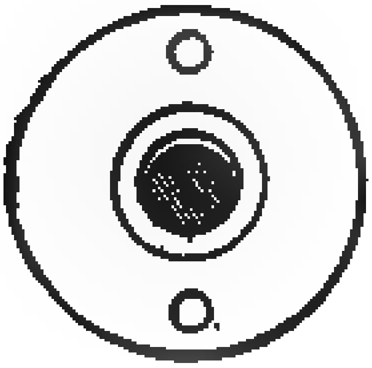
should be investigated immediately and its exact cause ascertained as, if the piston be striking the cylinder head, it not only may wreck the engine but cause loss of life.

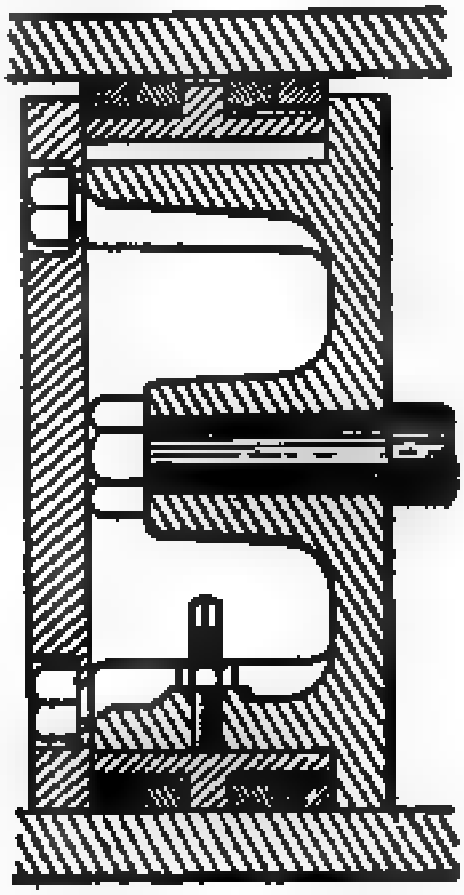
Engine Out of Line.—A great many knocks may exist on account of an engine being out of line. The shaft may not be at right angles with the cylinder; the latter may not be parallel with the guides, or the crank pin may not be parallel with the shaft, etc. Any imperfection in alignment will cause a knock, or heating of some part.

If the shaft be out of line with the cylinder, as shown in fig. 1,032, the thrust from the connecting rod will not bear evenly on the crank pin.

On the forward stroke, the crank brasses will bear sidewise against the face of the crank, while on the return stroke this side thrust will come on the head of the crank pin. Therefore, if the brasses have any side clearance, a knock will result from the side slap of the rod.

If the shaft be not at right angles to the engine, the connecting rod will be out of line with the cylinder and guides. This amount is greatest at the centers, or where the pressures are reversed, consequently the evil that





Figs. 1,033 to 1,035.—Effect of worn piston. An engine piston should occupy a perfectly central position within the cylinder for three principal reasons. First, when the bull ring gets down, as it is called, the upward pressure against the sections of the rings on top of the bull ring is greatly reduced and therefore the joint between the rings and the bore of the cylinder, or the cylinder wall, is not so tight and steam can more easily leak through to the exhaust side of the piston. This steam, of course, is lost for it escapes into the exhaust pipe without doing any work. The second reason is, that when the piston becomes worn and the center lies below the axis, or the horizontal center of the cylinder, the piston rod will not occupy a central position through the stuffing box. This condition always causes trouble and when the cause is not speedily removed, it often leads to expensive repairs. As shown in the above figure, when the piston is "down" the space under the rod occupied by the packing in the stuffing box is considerably reduced. The packing is compressed very tightly, when expanded by the heat, and the tendency is to wear the bottom of the piston rod faster than the top; in other words, the rod will soon be worn out of round. The packing above the rod will be too loose and will leak steam unless the gland is screwed up much more tightly than it should be under the proper conditions. This, it will be seen, wears out the packing much faster than need be and hence causes a waste of packing and incidentally a waste of oil, in the endeavor to keep the packing pliable and to keep the rod from becoming overheated. It will be understood that after the rod has once become out of round, the packing must be kept very pliable, even after the rod has been restored to its proper position, and it must be screwed up tightly in order to adapt it to the irregular shape of the rod. Thus the waste of packing and of oil will go on until a new piston rod is put in which, it is perhaps unnecessary to say, is an expensive undertaking in an engine of any considerable size. If the piston be much out of center, the rod will bear on the gland, as shown in fig. 1,033, the effect of which is the same as that just described except that it is much more rapid and harmful. The third reason is, that when the bull ring and the spider become out of center, the packing rings being pushed upward by the springs under them have a greater leverage upon the side of the grooves, as shown in fig. 1,035, and in a comparatively short time will tend to widen the grooves near the top. This gives the rings some lost motion which results in a clicking noise when the direction of the piston is reversed at each end of the stroke. The spider and follower plate in pistons fitted with a bull ring do not as a rule fit the cylinder as tightly as the body of a piston of the solid type having rings sprung into the grooves, so that a follower piston, as it is frequently called, has more leeway, or more room in which to move up and down in the cylinder.

results is greatest. The effect of the connecting rod being out of line with the guides is a tendency to produce a knock from the cross head being forced sidewise in opposite directions at each center and also to cause side knocking of the connecting rod in the cross head. The reason for this will be understood by referring to fig. 1,032, where A B, represents the center line of the cylinder, and C D, the path of the crank pin, with the shaft out of line.

When the cylinder is not in line with the guides sidewise there is a tendency to knock at the cross head on the head end of the stroke. Again if the piston be not snug fitting in the cylinder bore, it will slam against the

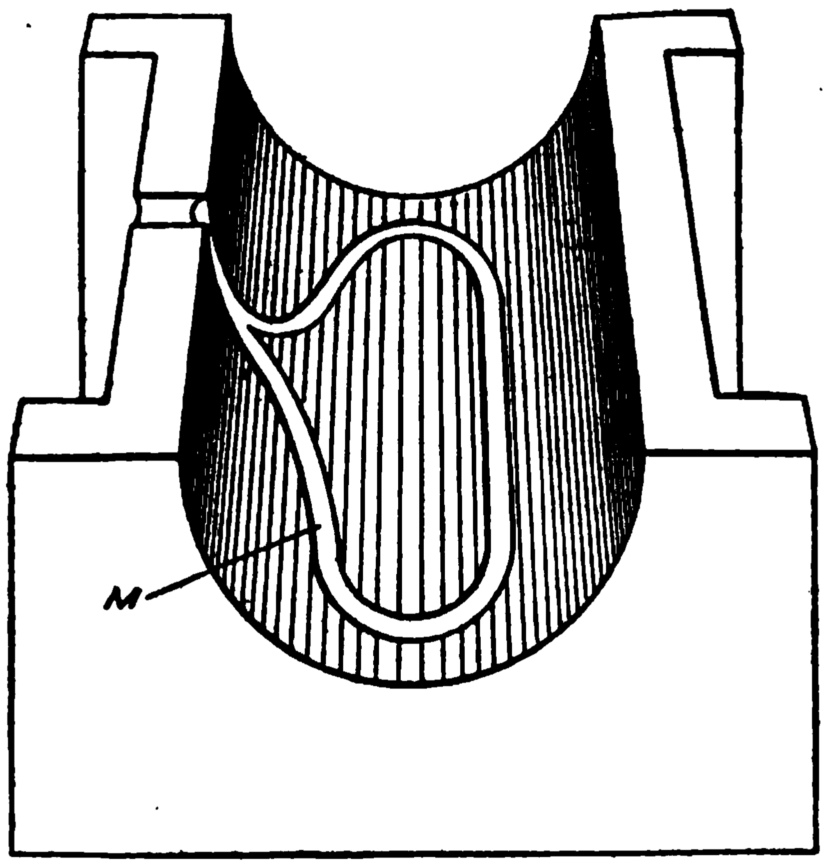


FIG. 1,036.—One method of grooving a journal to secure uniform distribution of the oil.

side of the cylinder and produce a knock. This is especially true in vertical engines, where the piston is unstable and responds to even a small sidewise force. The reason that there will be no knock at the head end in this case is that the cylinder is in line at that end. This is illustrated in fig. 1,032, where the cylinder not in alignment is shown in dotted outline.

Another effect of a cylinder being out of line is excessive wear on the piston and cylinder. When the engine runs in this condition the piston has a slightly rocking motion, the effect of which is to wear the piston most on its edges and destroy its true cylindrical form.

If the crank pin be not parallel with the shaft, the crank pin brasses will knock sidewise, and it will be impossible to take up the crank brasses much without heating; this condition will also cause a side knock at the cross head pin. If there be any clearance, the cross head end of the connecting rod will be thrown from side to side.

The crank pin can be tested for parallelism with the shaft by disconnecting the rod at the cross head, and noting any side movement of the free end of the rod. In making this test the crank pin brasses should be closely adjusted.

There will be a side knock at the wrist pin if the latter be not at right angles with the piston rod. To test the alignment of these parts, the connecting rod is disconnected at the crank end and the cross head brasses tightened. The position of the free end of the connecting rod will show the truth of the alignment, provided the piston rod be in line.

FIG. 1,037.—One method of testing the alignment of piston. In the end of the piston rod will be found the hole drilled for the lathe center when the rod was turned. This hole is exactly in the center of the rod and therefore it also represents the center of the piston. In this case a short, straight piece of steel rod or wire will be found useful. This is to be driven into the center hole as shown. Never attempt to take measurements from the faces of the nut, because it frequently happens that the nut is not finished, and the sides are therefore apt to be of different thicknesses which would spoil the work of centering, also the piston later on.

The wrist pin wears flat on both sides as shown in fig. 1,038. Provision is made on some engines for giving the pin a quarter turn thus presenting new surfaces for wear.

The crank pin wears flat on one side only. The reason for this is because the greatest pressure on the pin occurs while it is turning from the dead center to about one-quarter, or one-third stroke. Since the pin revolves, the same surface always receives this pressure, thus producing uneven wear or a "flat" as shown at A, fig. 1,039.

Connecting rod brasses sometimes in becoming hot close up and grip the pin, becoming distorted as shown in fig. 1,040. The manner in which brasses tend to close up is as follows: When the surface of the brass next

to the pin suddenly becomes hot, the brass tends to open as the result of expansion being greatest on the inside. The rigid strap or rod prevents any movement, and the metal is given a permanent set, then when the brass cools off the ends come together. After the brasses have become hot,

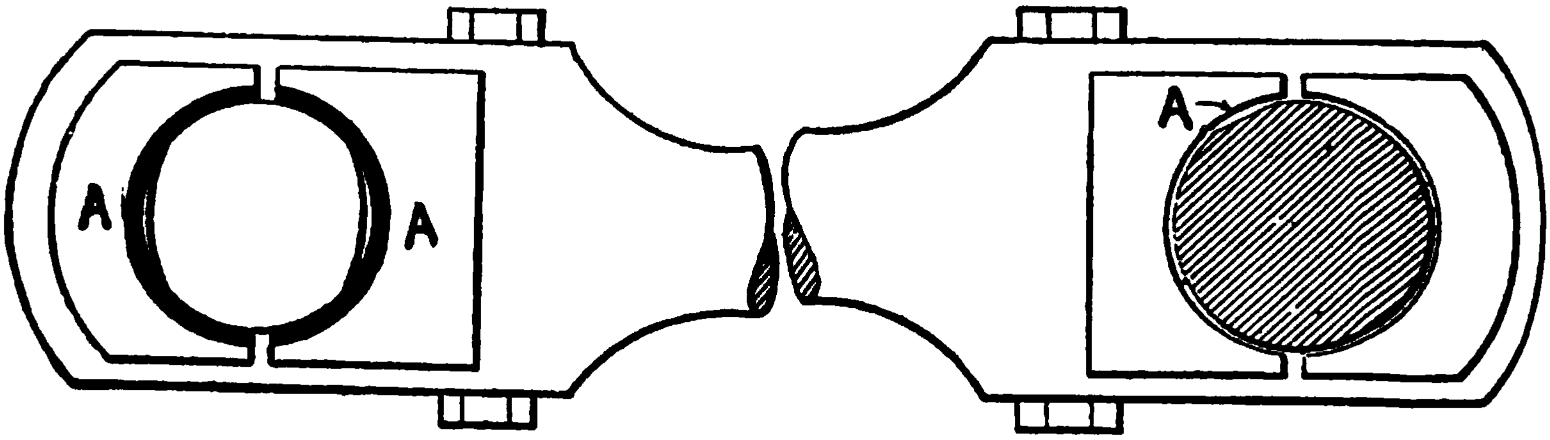


FIG. 1,038.—The uneven wear of the wrist pin.

FIG. 1,039.—The crank pin wears flat on one side only, as at A, because, due to its revolution, the same surface always receives the thrust.

it is usually necessary to relieve them on the sides where they hug the pin as shown in the figure. They are thus left more or less free to rock in the end of the connecting rod which causes knocks. The outer surfaces of the brasses may be machined true and liners inserted to secure the proper adjustment.

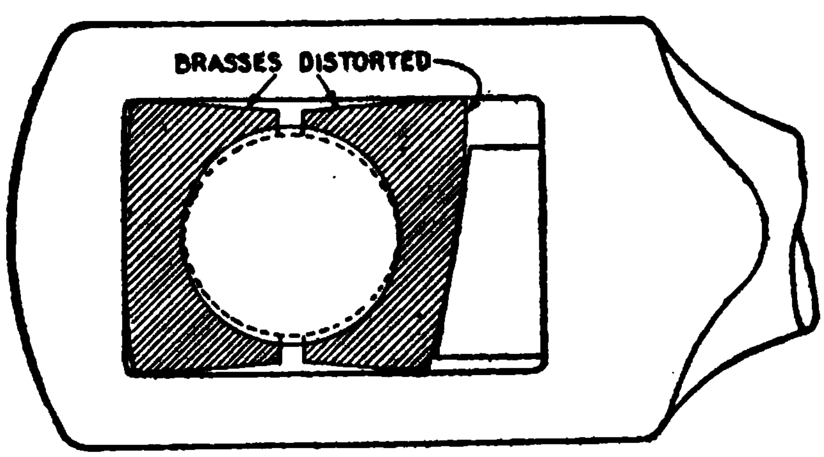


FIG. 1,040.—Gripping of the connecting rod brasses due to uneven expansion in heating.

In making connecting rod adjustments, it should be noted that while one brass remains stationary the other is moved toward it by two pins, keying up tends to make the rod longer; when the wedges are placed outside keying up tends to make the rod shorter. Again, when one wedge is inside, and the other outside, the length would remain constant provided

the wear was the same on both pins. However, since the greatest wear occurs on the crank pin, keying up the connecting rod will slightly change the clearance in the cylinder.

The continued adjustment of the connecting rod may change this clearance so much as to cause, 1, the piston to touch the cylinder head, or 2, the piston ring not to overtravel the bore at one end. In the first instance, the cylinder head may be cracked, and in the second, wear would eventually cause the formation of a shoulder with a resulting knock.*

Any inequality of piston clearance thus produced should be connected by cross head adjustment, or in case the piston is keyed to the cross head, the connecting rod length may be changed by means of liners.

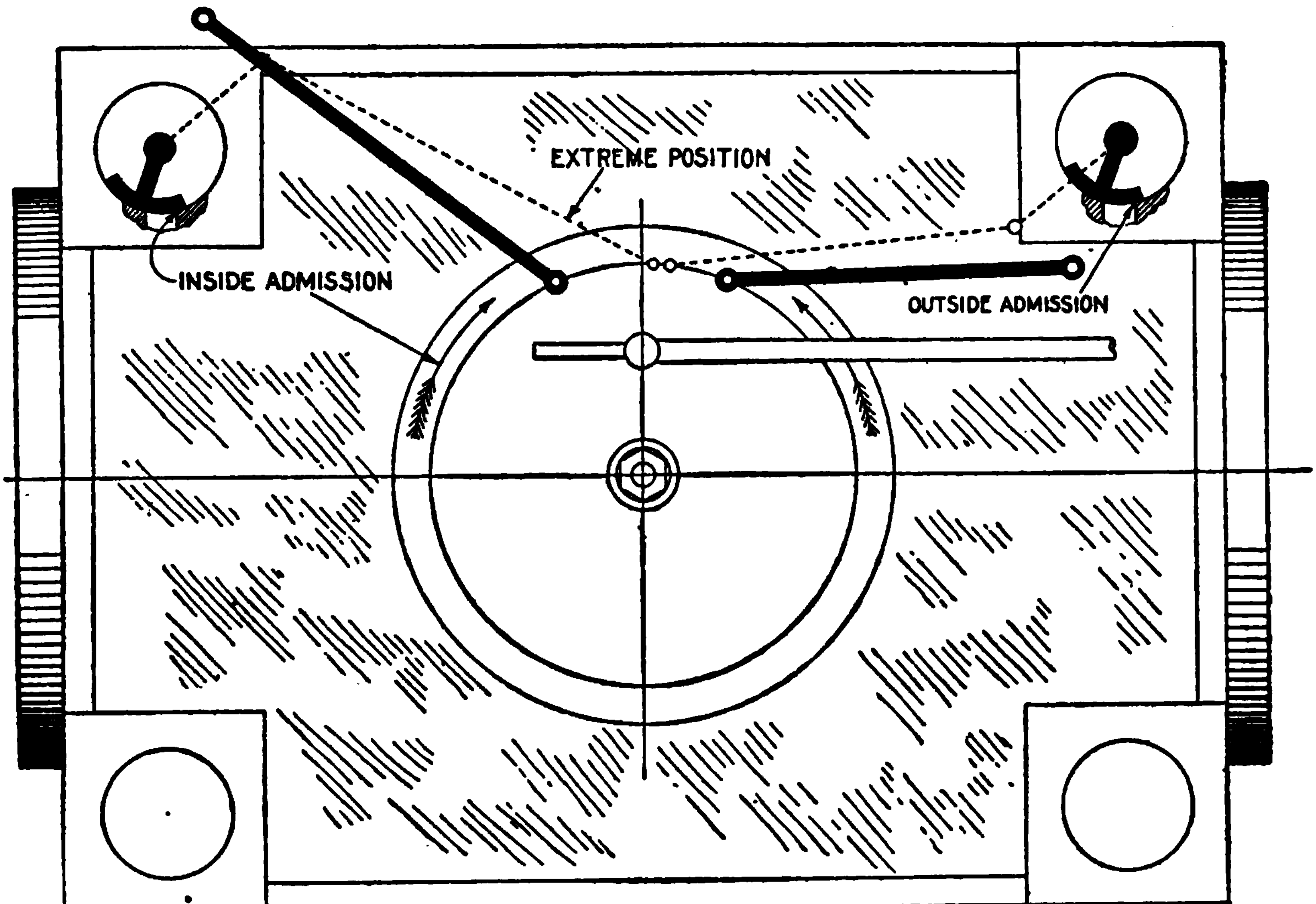


FIG. 1,041.—*Peculiarities of the Corliss gear.* 1, hook gear opening steam valves ***outward,*** gives ***inside admission;*** 2, hook gear opening steam valves ***inward,*** gives ***outside admission.***

Peculiarities of the Corliss Gear.—Where there are no stops to prevent the wrist plate being turned beyond its extreme positions, the engineer in using the starting bar should be careful not to turn the plate further than these points, as in so doing, the action of the gear may be changed or the gear itself damaged.

*NOTE.—In the design of a steam engine, every part having a reciprocating motion is made to overtravel any part against which it rubs so that the latter will wear uniformly without leaving a projecting surface or *shoulder* at the ends.

For instance, if, in operating an oval arm gear by hand, the wrist plate be turned so far as to bring the admission arm and steam rod in line, that is, on a dead center, the weight of the two parts will carry them downward,

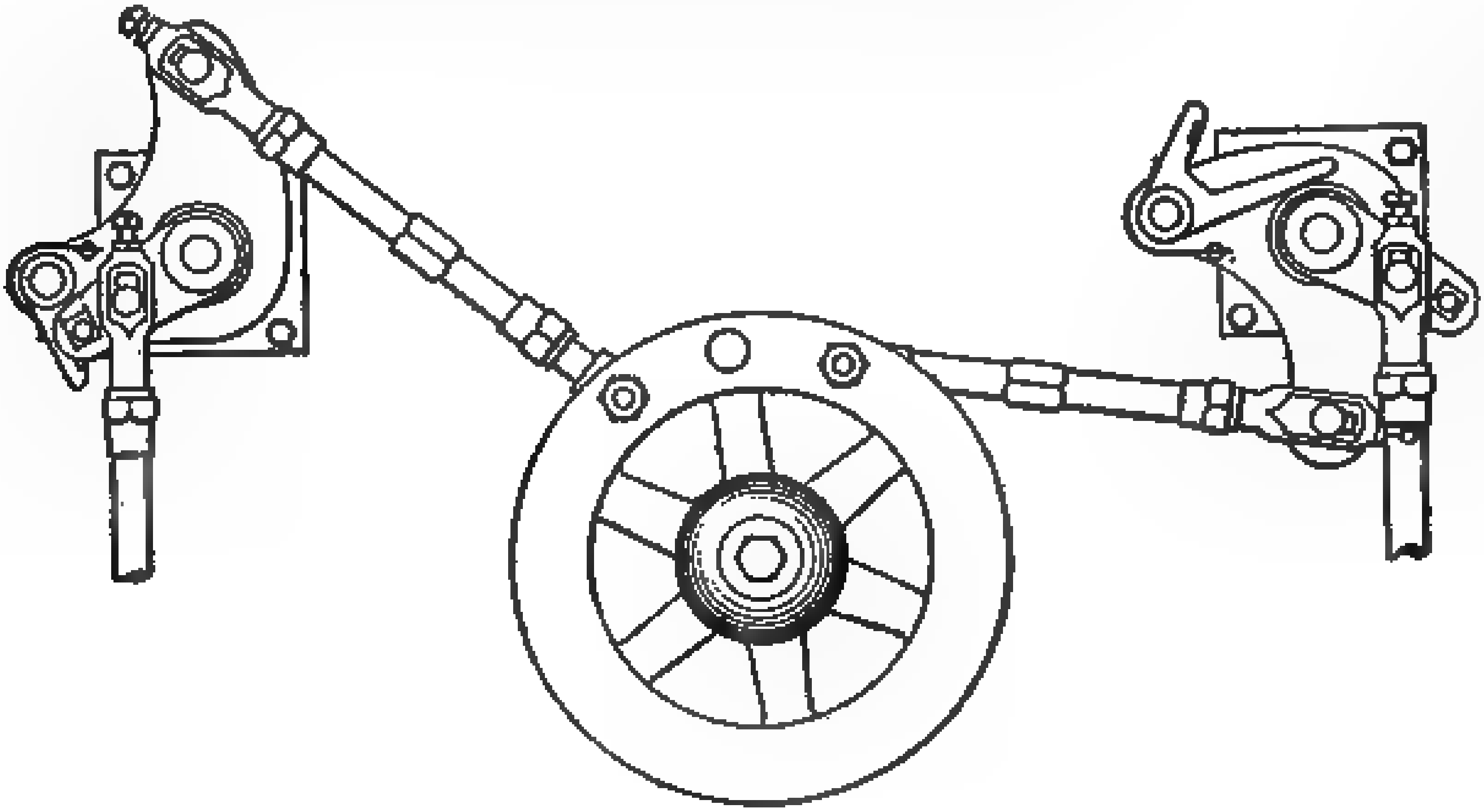


FIG. 1,042.—Peculiarities of the Corliss gear, showing steam valve mechanism turned over. This can happen if the wrist plate be turned too far in using the starting bar.

FIG. 1,043.—Sectional view of Filer and Stowell Corliss cylinder, showing double ported admission and exhaust valves.

causing the rotary motion of the admission arm to continue in the same direction as shown in fig. 1,042, instead of reversing when the wrist plate makes the return movement. Besides turning the steam valve entirely off its seat, the end of the steam rod in the absence of clearance would strike the dash pot rod, resulting in damage, should the carrier rod be hooked to the wrist stud during the stroke.

Again with any Corliss gear, too much movement of the wrist plate will similarly reverse the exhaust connections as shown in fig. 1,043. This will cause the exhaust valve to remain open, and allow steam to blow through to the exhaust pipe.

FIG. 1,044.—Exhaust valve turned over by too much movement of the wrist plate in using the starting bar. If the right hand bell crank be moved back to the position in which the left hand one appears, the engine will be in proper adjustment provided the wrist plate was not thrown back far enough to turn one of the exhaust valves over, as indicated by the right hand jim crank as here illustrated. In one case an electric lighting engine was started in this condition and run for some time before it was discovered by the engineer. This is possible because the exhaust valve would remain open, and while this would allow steam to blow through to the exhaust chest and be wasted, it would not offer resistance to the advance of the piston, as it would in case the key which fastens the jim crank to the valve stem would become loosened, work out and drop down on the floor, leaving the exhaust valve shut. In that case compression would rise high enough to be a serious matter, for it would probably stop the engine. In more than one case the engineer has found his Corliss engine slowing down from this cause; but by quickly inserting the key as the jim crank came into proper position for it, normal speed has been restored and nobody else knew what caused the variation.

The valve gear may be damaged if the exhaust arm be reversed by the latter, in its travel, coming in contact with some other part of the engine.

If the key which secures the exhaust arm to the valve stem should become loosened and fall out, the exhaust valve might remain shut or partially so, probably causing the engine to stop, or slow down owing to the excessive compression, or loss of pressure thus caused. If, therefore, the engineer find his engine slowing down, he should examine the exhaust arms as a possible cause of trouble. Sometimes a key is used to take up

the lost motion of the exhaust rod as in fig. 1,045, and as the parts continue to wear, the repeated adjustments finally cause the end of the key to strike the valve stem bearing, throwing it up again, and allowing the brasses to remain loose. This is an instance, like many others, where the cause of the trouble is not readily discovered.

The crab claw gear will sometimes cause the engine to race, taking steam full stroke for a revolution or two, due to wear of the dies. The wearing of the contact edges, and the consequent grinding of the same, changes the relative position of the parts

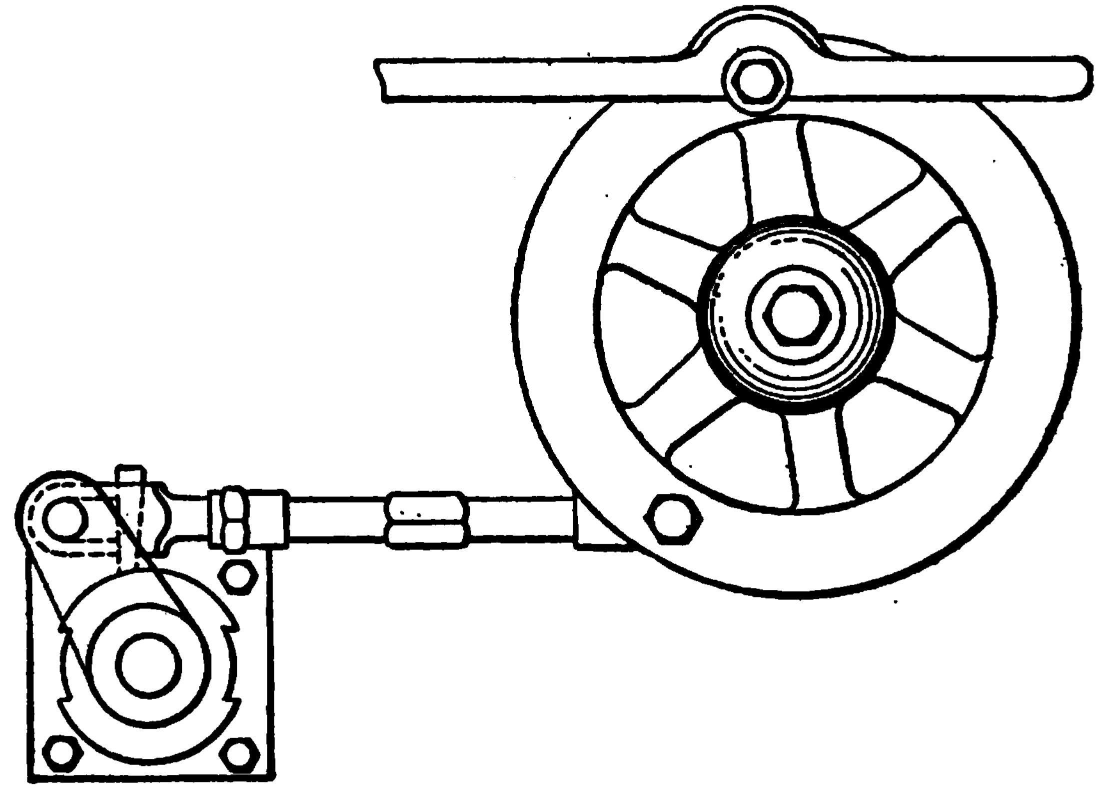


FIG. 1,045.—Adjustment key too long on exhaust rod, causing the brasses to remain loose on account of striking the valve stem bearing. Such a fault is not always readily discovered unless the engineer's attention is called to it by the experience of others along the same line, and although a remedy can be provided by cutting a piece off from the key, it is much better to put a liner behind the box and thus raise the key until it no longer rubs on the stationary bearing.

to such an extent that the knock off cam cannot release the hook if the load be such as to require a late cut off.

The reason for this is that the shortening of the dies changes the position of the curved limb of the crab claw with respect to the knock off cam. That portion where contact with the cam occurs at late cut off being lowered, the increased clearance prevents the cam striking the crab claw limb at this part. Hence, if the engine be heavily loaded, requiring a late cut off, it will take steam full stroke for one or two revolutions, making it race.

To determine the extent of wear, calipers are placed in the center of the die stud, and the outline of the circular limb of the crab claw followed with the other end. If the calipers run off instead of following the curve of the claw, and describe some arc, the stud die is moved along the spindle until the calipers will follow the curve of the claw. The distance between the two dies then indicates how much longer a new die should be.

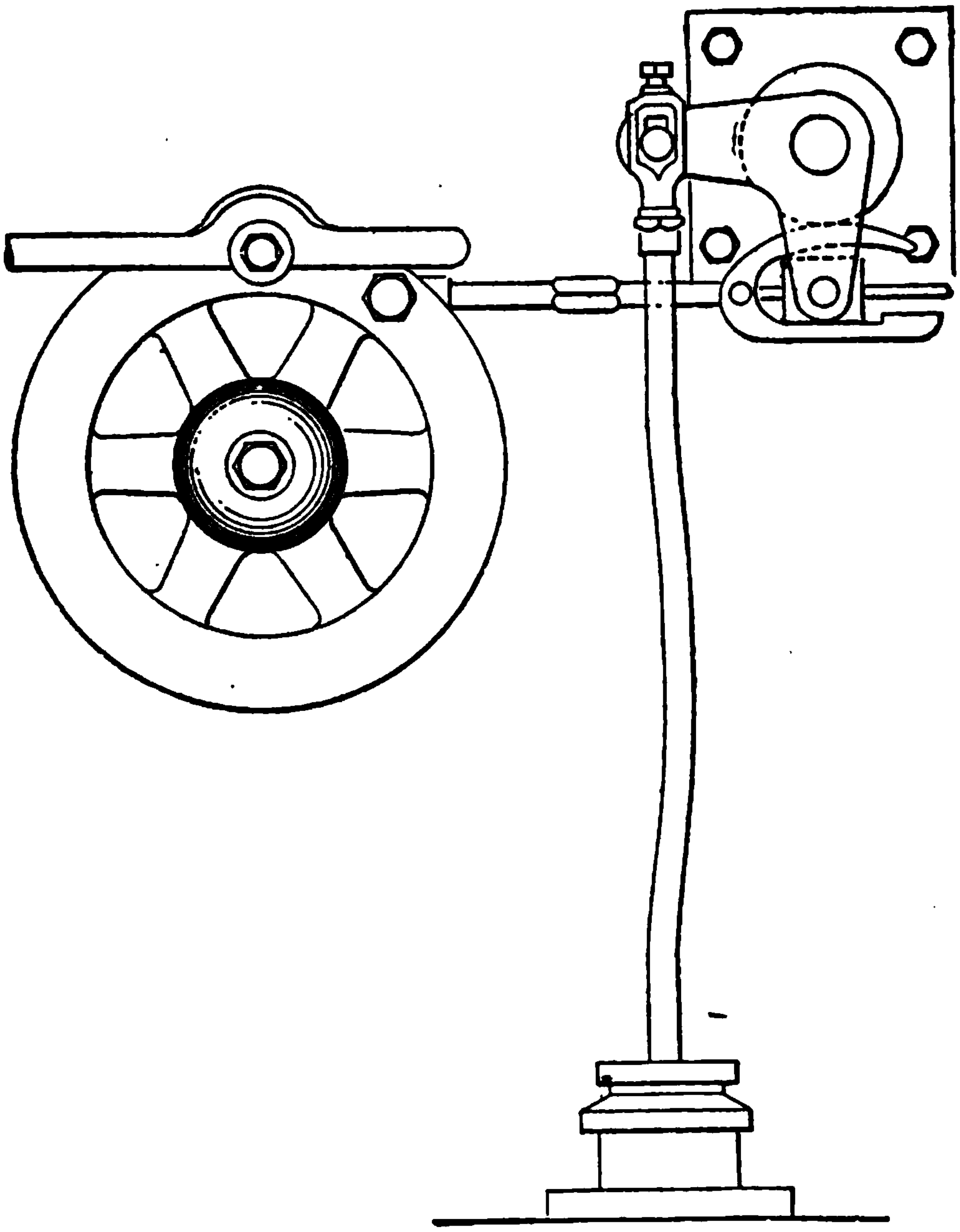


FIG. 1,046.—Dash pot rods too long. Before starting a Corliss engine, the engineer should make sure these rods are correctly adjusted, by moving the valve gear with the starting bar to the extreme positions, otherwise *the rods may be bent as shown, the bonnet castings broken, or both.*

The crab claw gear sometimes fails to cut off if the engineer neglect to properly lubricate the steam rods. In severe cases considerable damage may be caused.

The stud die may heat from lack of oil, and grip the steam rod making cut off by the dash pot impossible; the engine will then take steam full stroke, resulting in a burst fly wheel and wrecked building if not discovered in time.

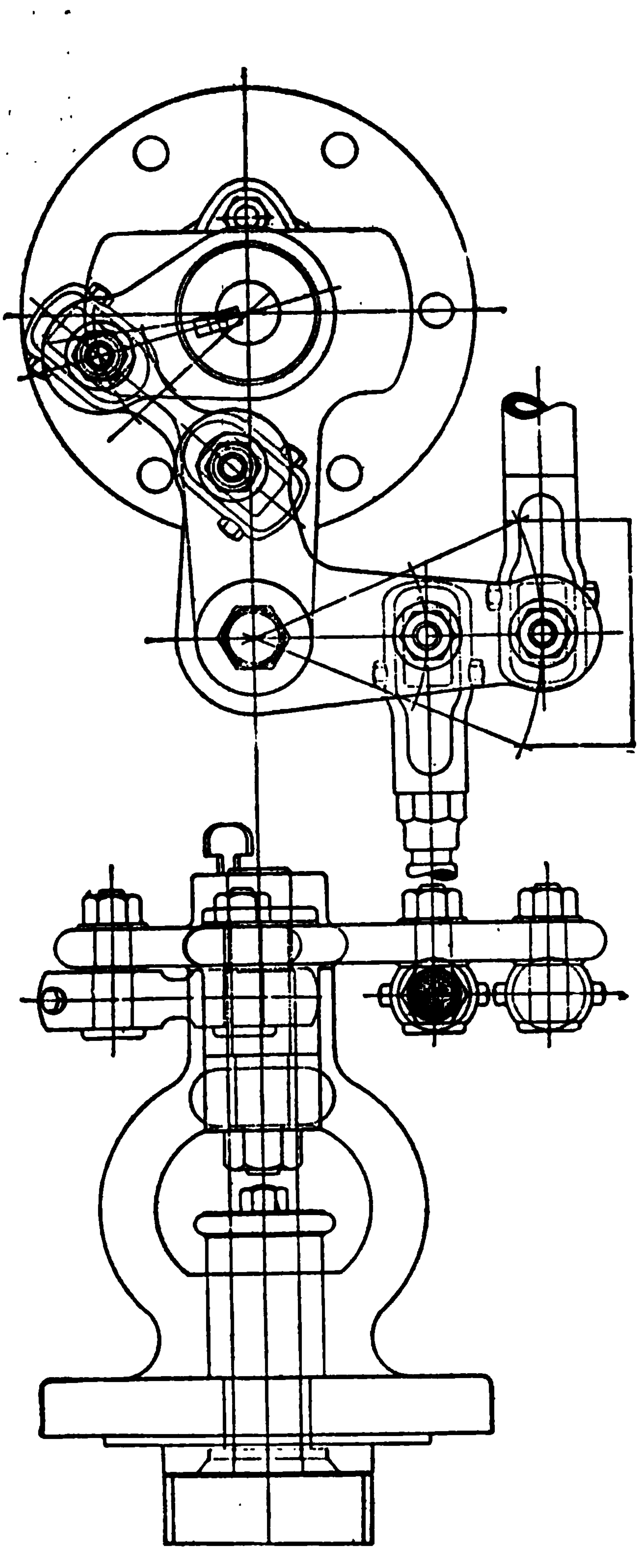


FIG. 1,047.—Rice and Sargent Corliss exhaust valve gear. The link is interposed between the valve rod bell crank and the exhaust lever to allow the valve to pause at the time the pressure upon it is heaviest, thus minimizing the friction loss and giving a rapid motion at the time of opening and closing the ports. Separate eccentrics are used on all Rice & Sargent engines for operating the steam and exhaust valves. This permits a long range cut off (80 per cent), and adjustment of compression and release not obtainable otherwise.

How to Stop a Condensing Engine.—When shutting down a condensing engine, the following principles should be kept in mind:

> At the instant the throttle is closed and steam has ceased to exhaust into the condenser, a higher vacuum will form which will increase the rate of flow of the cooling water through the injection valve. The cooling water will then enter the condenser faster than the pump can remove it, in which case flooding will result with damage to the engine.

Therefore, unless the throttle and injection valves are so placed that they may be operated simultaneously, the engineer in stopping the engine should first reduce the supply of cooling water. The injection valve is turned to reduce the

FIG. 1,048.—Vilter simple girder frame Corliss engine; crank side. The girder is so designed that the maximum depth comes near the crank end of the girder.

supply, until the vacuum falls from, say, 24 inches to 10 inches. The throttle may now be closed, and then the injection valve. *The air pump should never be stopped until the engine has ceased to move.*

This rids the condenser and pipes of water, and is a measure of safety in case the injection valve be accidentally left partially open.

When the air pump is attached to the engine the danger of getting water into the cylinder is greater, because the pump slows down with the engine. With this arrangement, the injection valve should be near the throttle where both may be closed at the same time, or if otherwise placed, the injection valve should be nearly closed before shutting off steam.

All jet condensers should be fitted with a relief pipe and valve opening to the atmosphere.

This valve should be within easy reach of the engineer. After the throttle and injection valve are closed, the relief valve should be opened to break the vacuum and thus guard against water reaching the cylinder.

FIG. 1,049.—Vilter simple girder frames Corliss engine; valve gear side.

CHAPTER 16

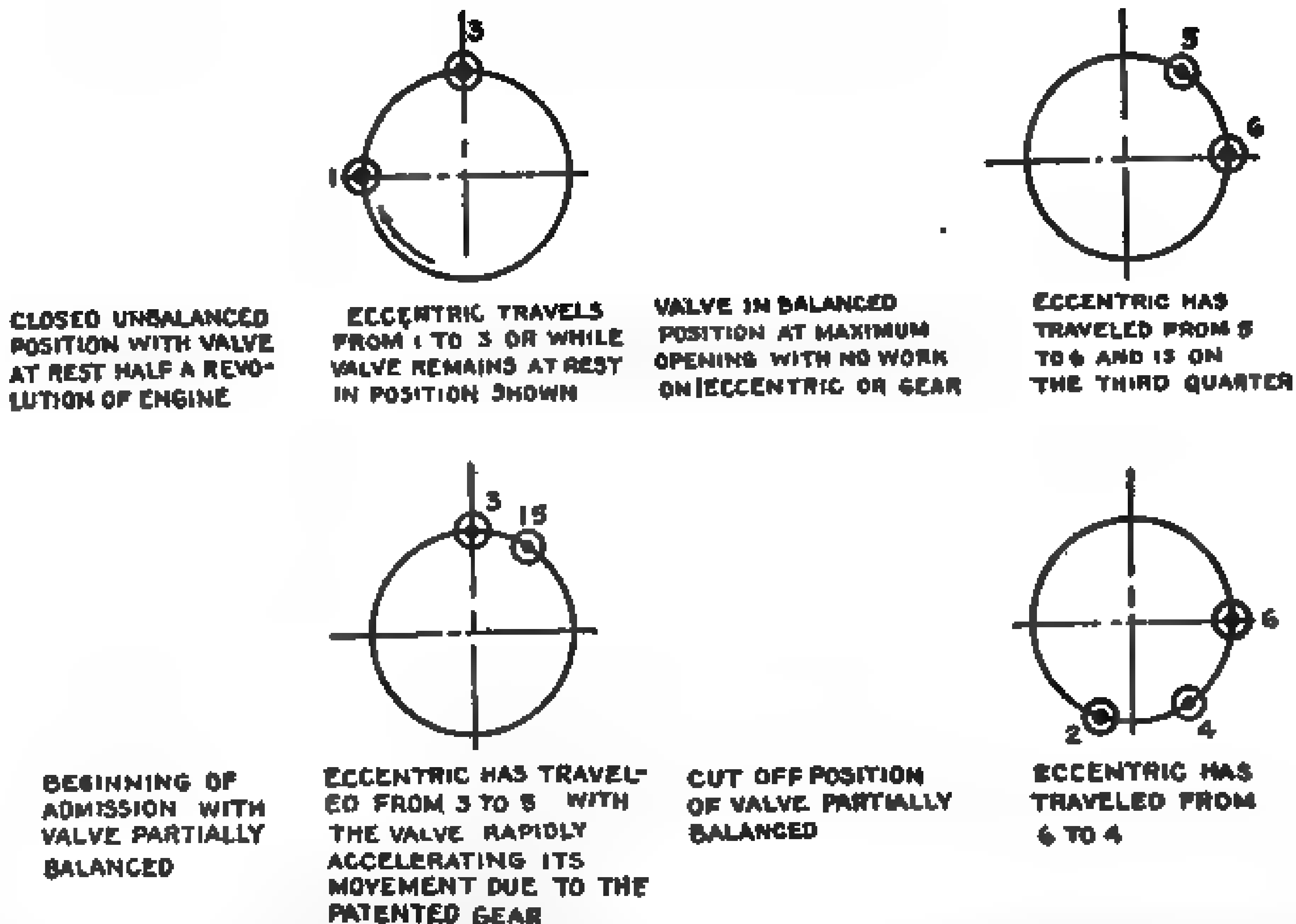
FOUR VALVE NON-RELEASING ENGINES

The rapid increase in the use of electricity for power and lighting purposes has caused an insistent demand from the purchaser for a more economical engine than the ordinary high speed automatic type. When the advantages to be derived from connecting the electric generator direct to the engine shaft began to be generally recognized, it was found that the Corliss engine, owing to its limited rotative speed was not well suited. This resulted in the development of a new class of engine, possessing many of the good features of the Corliss, but capable of higher rotative speed on account of having a *non-releasing* valve gear. Some of these engines are quite similar to the Corliss, even to wrist plates, steam and exhaust rods, and may be easily mistaken for such, but it should here be mentioned that an engine with a non-releasing valve gear is not a Corliss, *although some of the makers have traded on the name of* ***Corliss*** in advertising.

Rocking valves were used before Geo. H. Corliss' time, the particular features which distinguished his engine being the releasing gear, and dash pots. Each type of engine is excellent —for the particular service to which it is adapted.

The drop cut off, while a most excellent feature with respect to the steam distribution, has the objection that it does not admit of high rotative speed on account of the lag due to the inertia of the drop mechanism, wear, etc. The Corliss engine therefore requires a large space in proportion to the power developed, and since the governor controls the speed

FIG. 1,050.—Diagram of Ideal accelerating non-releasing valve gear, showing progressively the movements of the gear for one end of the cylinder. The eccentric E, travels around its circle and drives the eccentric rod, which connects to the wrist plate pin W. Inside the gear case is a rocking plate with two arms, A and A′, for driving the head end and crank end valve linkages. To the arm A, is pivoted a lever on which is formed a cylindrical tail rod, which slips through a trunnioned guide C, which serves as the fulcrum. At a suitable point B, on the lever is pivoted a link which connects to the valve crank D. For the sake of clearness, the diagram omits the connecting rods which reach from gear box to cylinder, as these rods and their connecting arms merely transmit the movements without affecting the velocity ratios. The corresponding positions of the various points are numbered alike. Starting at the position 1, as the eccentric moves up to position 3, the arm A, moves from its position 1, to position 3, and the point B, on the lever is forced to the left to its position 3, shoving the tail rod R, through the trunnioned guide C. During this movement, the link connecting points the B and D, swings about D, as a center, so that the valve crank is not moving during this period. As the eccentric moves from 3 to 5, point W, travels from 3 to 5 on its arc, and point A, travels downward from 3 to 5, point B, being drawn downward to the right, and tail rod R, sliding to the right through the trunnioned guide C, as the point B, advances from point 3, on its curve to point 5, on its curve. This draws the valve crank downward from its position 3, to position 5, and the valve is rotated by the amount of its lap, bringing it to the point of admission. As the eccentric travels from 5, to 6, the points W, A, B, and D, make the corresponding travel, and the valve opens for the admission of steam. As the eccentric travels from 6, to 4, point W, travels back from 6, to 4, on its arc, and points A, B and D, make corresponding movements, bringing the valve back to the point of cut off; and when the eccentric reaches the position 4, the valve is brought to its full closed position. While the eccentric moves from position 2 to position 3, or approximately half a revolution of the engine, the valve remains at rest.



FIGS. 1,051 to 1,058.—Unbalanced and balanced positions of Ideal valve gear and corresponding eccentric positions. From 4 to 2, the eccentric moves the valve on the lap to dead rest closed position as in fig. 1,051, and from 2 to 1, the point of beginning, the valve is stationary, thus completing the cycle. It will be noticed that during the period while the valve is at rest it is unbalanced, that is, the steam pressure tends to force it against the seat, while from the time that the ports begin to admit steam until the valve cuts off, the steam pressure is balanced on the two sides of the valve. The high speed of valve obtained during opening and closing, is due to the fact that during this time the eccentric is on the quarters and is giving the maximum speed to the eccentric rod and connecting gear, and has the maximum moment to move the valves at the time when they call for the greatest amount of driving power.

Fig. 1,059. The Lane and Bodl ppearance of the valve gear is similar to that of a Corliss, yet it differs from the latter in the at he cut off is positive, as the action of a movable rotating eccentric and es the angular advance are two wrist plates; the exhaust valves are eccentric. This valve single eccentric if fitted with one of the swingi tating type. The cut changing both the angular advance and travel.

of the engine by varying the point of cut off, the slow rotative speed does not admit of closest regulation. In order to overcome these defects, the releasing gear has been displaced by a positive cut off in the design of the non-releasing gear, thus adapting it to the highest rotative speeds.

With the positive cut off there is no inertia lag, and the valve closes while the crank moves through a definite angle no matter how great the rotative speed. While this angle may be somewhat greater than with the Corliss gear at slow speed, it is, for high

speed, less than would be possible with the latter if run at an equal number of revolutions. In order to make the positive cut off as quick as possible double and sometimes triple ported valves are used and given liberal travel. The numerous non-releasing cut off engines may be classified:

1. With respect to the valves, according as they have,

   a. Semi-rotary valves; or
   b. Sliding valves.

2. With respect to the manner of cut off, according as they have.

   a. Direct cut off; or
   b. Riding cut off.

3. With respect to the method of variable cut off, as

   a. By variable angular advance;
   b. By combined variable throw and variable angular advance.

Rocking Valves, Direct Cut Off.—Fig. 1,059 shows a four valve shaft governor engine with rocking valves, and direct cut off. This engine strongly resembles a Corliss, even to wrist plate, and steam and exhaust rods; in fact it might easily be mistaken for such at first sight. On closer examination, however, the absence of dash pots and releasing gear will be noted.

An operating arm is keyed to each valve stem, and pivoted to one of the rods which transmit motion from the wrist plate.

The engine has double eccentrics, hence there are two wrist plates, although for light duty, all four valves could be operated from one wrist plate by a single eccentric.

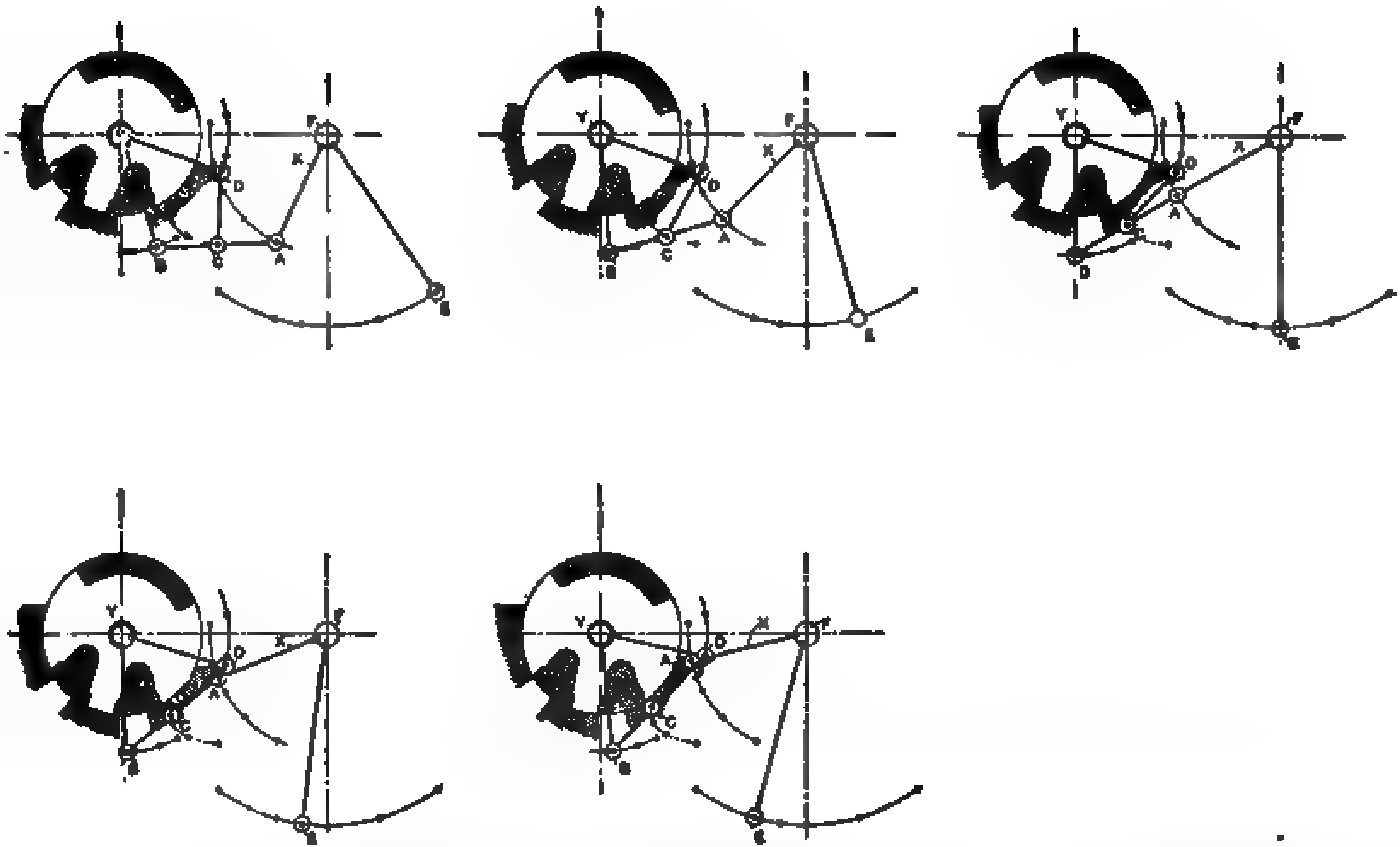
The valves are of the double ported type, being similar in construction to the regular Corliss valve. A variable cut off is secured by the method of ***combined variable throw and variable angular advance.*** (See page 244). This is done automatically by the movable steam eccentric controlled by a shaft governor. The exhaust valves are operated by means of a separate eccentric having a fixed position on the shaft.

Rocking Valves in Cylinder Heads.—Sometimes in order to reduce clearance, engines are designed with the valves in the cylinder heads as shown in fig. 1,060. The steam valves of this engine are operated by the Armstrong non-releasing gear, as shown in figs. 1,061 to 1,066, and in fig. 1,068. It opens and

FIG. 1,060.—The Ball four valve engine, with valves in the cylinder heads. The object of this arrangement is to reduce clearance. The valves are operated by the Armstrong non-releasing gear as shown in fig. 1,068 and diagrams 1,061 to 1,066.

closes the valves at the proper times, and in the remainder of the time, or about half of the revolution, the valves are at rest without being released.

The valves have their highest speed at the instants of opening and closing. The movement of the valve begins in the middle of the valve gear travel, when all parts of the gears are at their highest speed. The gear is enclosed in a case which is partly filled with oil. Figs. 1,061 to 1,066 show the valve gear in elementary form plotted in several positions.



FIGS. 1,061 to 1,066.—Diagrams showing operation of the Armstrong non-releasing valve gear. From the diagrams it will be seen that the link C D, connected to the valve crank, swings around the pin D, without imparting any motion to the valve crank Y D, one-half of the total movement of the eccentric rod which drives it. At usual loads the valves have their highest speeds at the instant of opening and closing. ***In construction,*** in smaller cylinders the valve seats are formed integral with the cylinder, as in the usual type of long stroke Corliss. As these cylinders usually are longer in proportion to their diameter than the larger ones, this keeps the clearance sufficiently low to permit this construction. The admission valves of these smaller cylinders are double ported and the exhaust valves single ported. In these cylinders the exhaust valves are driven direct, no wrist plate being necessary.

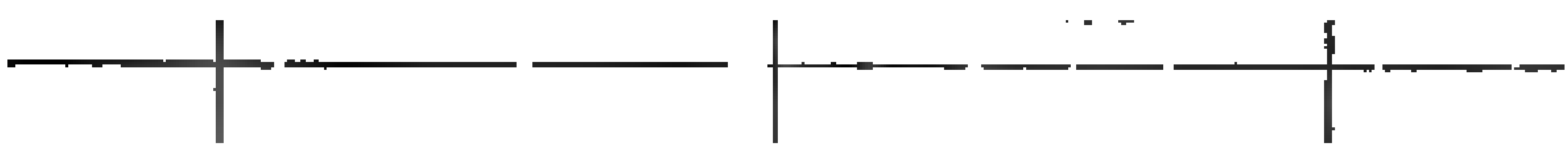


FIG. 1,067.—Diagram of valve movement of Ajax engine. The cylinder has attached to one side two steam exhaust valve brackets operated from independent swing plates attached to separate eccentrics. The exhaust eccentric is fixed in its throw but can be varied as to the angular position with reference to the crank. The steam eccentric is of variable throw and comprises a part of the Robb-Armstrong-Sweet governor described in another paragraph. The arms attached to the valve stems are connected to two swing plates which produce a long dwell when the valves are open and a corresponding condition when they are shut, but at the moment of opening and the moment of closing, the speeds of the valves are most rapid. Neither the steam nor exhaust valves are ever entirely at rest except at their moment of reversal. This condition prevents what is known as molecular interlocking, or a tendency for the surfaces to seize. The steam valves are practically balanced when they are moving toward the admission or closing points, producing a minimum of strain on the governor and valve connections.

Rocking Valves, Riding Cut Off.—An interesting example of the application of the riding cut off to semi-rotary valves is shown in figs. 1,069 and 1,070. The cut off valve rides on the inner circular surface of the steam valve, being shown in the figures in black section.

Each valve is operated by means of an arm keyed to the stem, motion being transmitted from the wrist plates by rods pivoted to both.

FIG. 1,068.—The Armstrong non-releasing valve gear as used on the Ball engine. The system of levers is such that it gives a quick movement to the valves in opening and closing, and allows them to remain at rest during the other periods. The cut off is changed by *combined variable travel and variable angular advance.*

There are two eccentrics and two wrist plates, but it should be noted that both steam and exhaust valves are operated by a fixed eccentric, and the cut off valves by a movable eccentric, free to rotate on the shaft giving variable cut off by variable angular advance as controlled by the shaft governor.

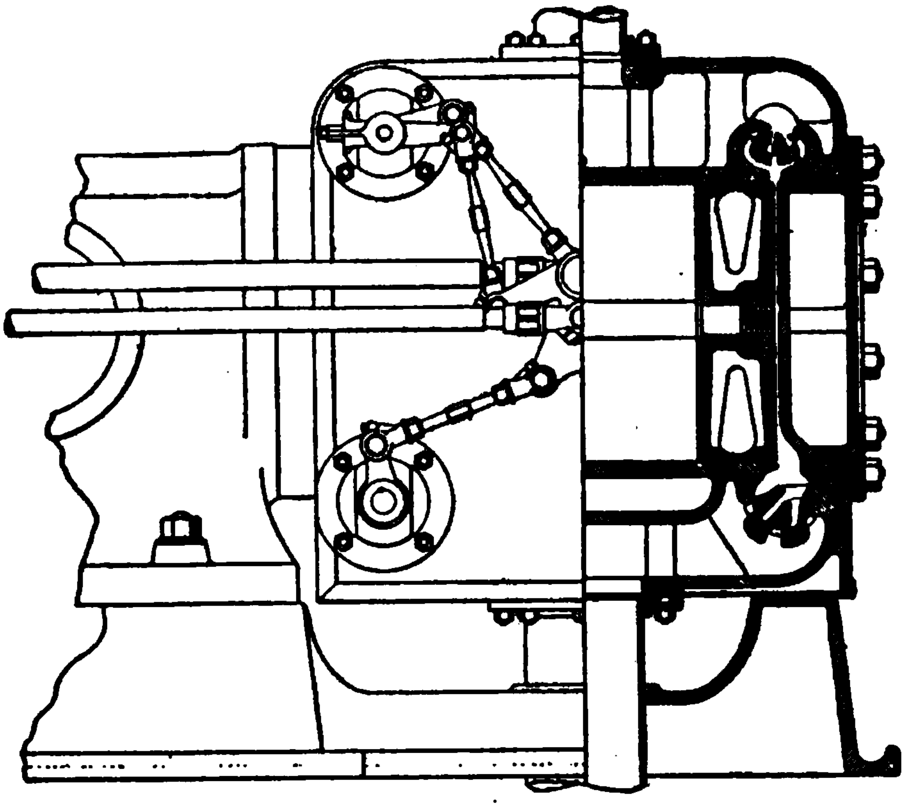


FIG. 1,069.—The Clark four valve riding cut off engine; there are in reality six **valves, each** steam valve consisting of a main admission valve, and a cut off valve. A fixed eccentric operates the main and exhaust valves, and a movable (rotating) eccentric (giving ***variable angular advance***), the riding valves, the cut off being controlled by a shaft governor.

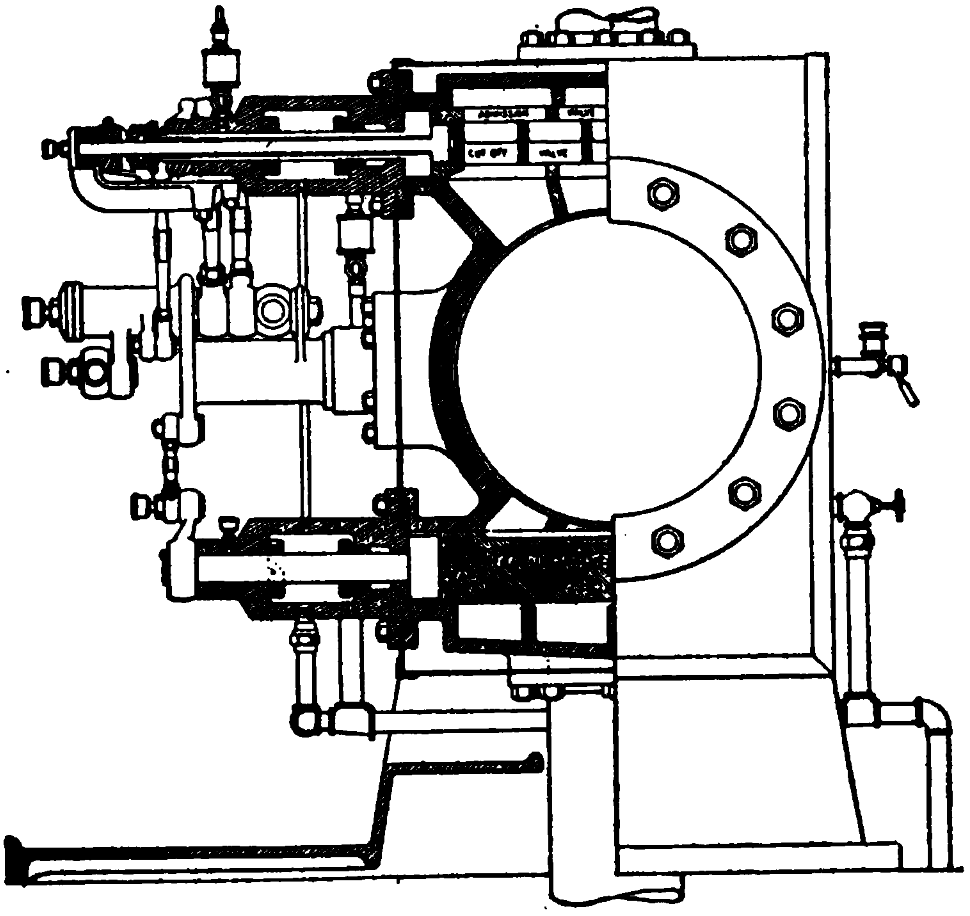


FIG. 1,070.—The Clark four valve riding cut off engine; end view showing **valves and** connections.

Sliding Valves, Direct Cut Off.—Flat instead of circular valves are often used on four valve engines. The engine

FIG. 1,071.—The Porter-Allen four valve high speed engine. Sliding pressure plate valves are used which give four openings for admission and exhaust, thus reducing wire drawing to a minimum. This engine is designed for the highest rotative speeds. The valves are operated by the Fink link motion, described on page 536.

illustrated in fig, 1,071, has valves of this type, and is constructed for extreme high rotary speed.

There are two independent admission valves, with direct cut off, and two exhaust valves. The cut off is varied by the Fink link,* which gives variable travel under control of the governor, both admission and exhaust valves being operated from one eccentric. The valves are balanced by adjustable pressure plates.

FIGS. 1,072 and 1,073.—Valve connections of the Porter-Allen engine, illustrating the differential valve movement. A wrist motion is introduced into the connection of the admission valves, as here shown, and which effects a useful modification of the movement of these valves.

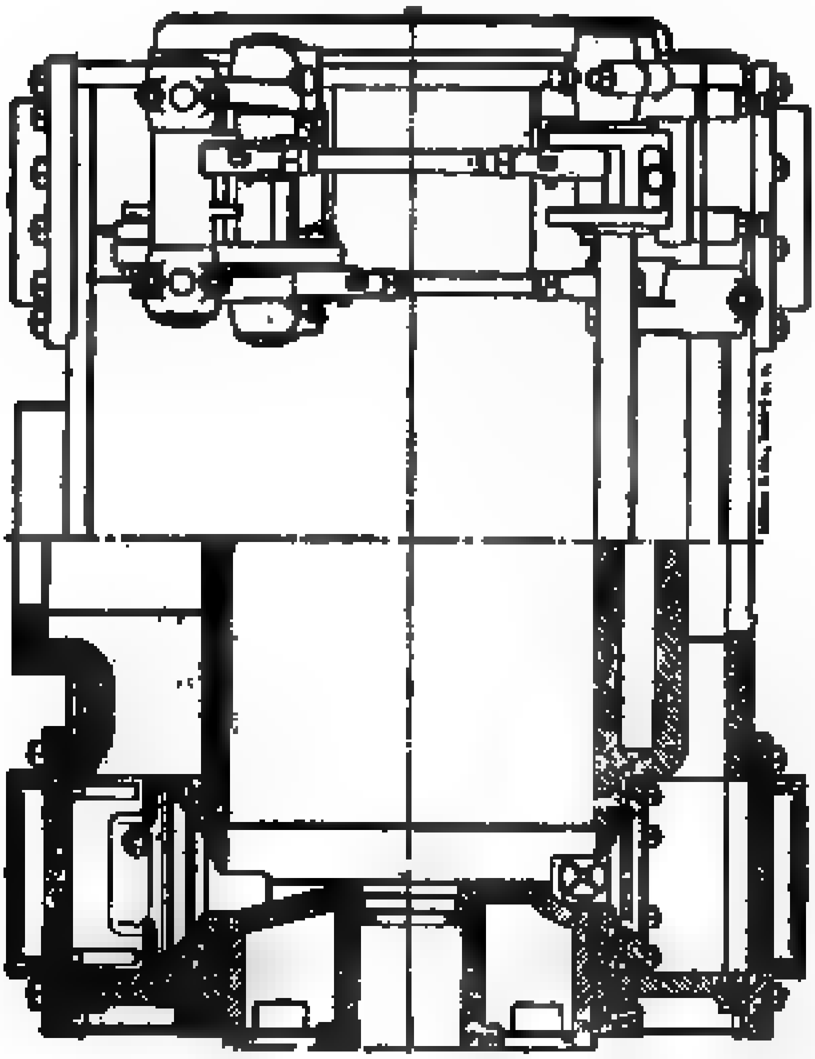
Figs. 1,072 and 1,073 show the valve connections in plan and elevation, and illustrate the differential valve movement.

A wrist motion is introduced into the connection of the admission valves, as shown, to modify their movements.

In this movement an arm, which is connected by a rod with the block in the link, communicates, through a rock shaft, motion to two other arms, causing them to vibrate in the same vertical plane in which the valves move.

*NOTE.—The operation of the Fink link is explained on page 336.

FIGS. ˙ cylinder head of McIntosh & Seymour type F engine. In fig. 1,074, the cylinder and head
are g. 1,075, all stripping necessary to remove the head has been performed This consists
of n l bolts for cylinder, steam and exhaust connections, which all have ground joints and
two Fig. 1,076 shows head completely free of cylinder. Taking off and putting back the cyl-
inde tting.



FIGS. 1,077 and 1,078.—McIntosh and Seymour four valve riding cut off engine, having valves of the gridiron type. The valves are unbalanced, and free to lift. The cut off is changed by *variable angular advance.*

Each of these arms alternately rises nearly to the vertical position, while the other, at the same time, descends to and beyond its dead point. Each, by a separate connection, imparts motion to one of the admission valves, and, at the top of its vibration, causes it to open and close its port swiftly, and then, descending to its idle arc, reduces the motion of the valve to an interval practically at rest. These movements can be followed in the cut, where the upper arm is about to move in its arc to the left, and thus, through the lower connection, to open the port at the further end of the cylinder, while the lower arm will be scarcely moving its valve at all. In this manner the width of the opening is increased chiefly by the difference in the length of the levers, while at the same time, about one-half of the lap, or the useless motion of each valve after it has covered its port, is avoided, so that smaller valves and narrower seats are employed, giving a liberal port opening with moderate travel.

Sliding Valves, Riding Cut Off.—An example of sliding valves with riding cut off is shown in figs. 1,077 and 1,078, in which valves are of the gridiron type. The engine here illustrated has a valve motion which is positive throughout. The flat gridiron valves are unbalanced and the port edges

FIG. 1,079.—McIntosh & Seymour (type F engine) cylinder head with lagging removed.

FIG. 1,080.—Method of removing McIntosh & Seymour (type F engine) valve. The valve together with its cover and bell crank lever can be removed by simply unbolting the ground joint of the cover and taking out one quick-removable pin. They can also be replaced without involving any readjustment of valve gear.

overtravel to prevent the formation of shoulders. Since the valves are multiported the travel is small, requiring only from one-half to one and one-half inches, according to the size of the cylinder.

The valve gear is simply an arrangement of links, rock shafts and slides for transmitting the motion of the eccentrics to the valves. The action of the gear is such as to distort the motion imparted by the eccentric, hastening the movement of the valve when near one end of its stroke and at the other end causing a pause in its motion, so that while a rapid opening and closing of the port is secured, the valve remains practically still while closed.

With this gear the cut offs at both ends of the cylinder are equalized at all loads.

The main valves are driven by a fixed eccentric controlling admission, release and compression. The riding cut off is operated by a movable eccentric, the cut off being controlled by a shaft governor which varies the angular advance of the movable eccentric.

CHAPTER 17

POPPET VALVE ENGINES

The chief feature of these engines is the type of valve used, which is entirely different in construction and operation from the slide or piston valve.

The word poppet is a variant of *puppet* which has several meanings, the mechanical definition being, *the head stock of a lathe;* the slight similarity in form between the pulley in the head-stock and the valve, probably accounting for the term being applied to the valve.

By definition, then, a puppet or **poppet valve** *is a valve which, in opening, is lifted bodily from its seat instead of being hinged at one side, or sliding over its seat.*

Poppet valves are largely used on steam engines in Europe. as are Corliss valves in this country, and are almost universally used on gas engines.

According to Prof. Furman, "the chief advantage of these valves is the absence of sliding motion in operation, which outweighs, in internal combustion engine work the disadvantages of its reciprocating motion at high speed, of warping when made in large sizes and when made in pairs on a single stem, and of noise when seating." It might be added that certain types are practically balanced with respect to the steam, the small excess pressure tending to keep the valve closed. Less movement is required than with other forms of valve for a given

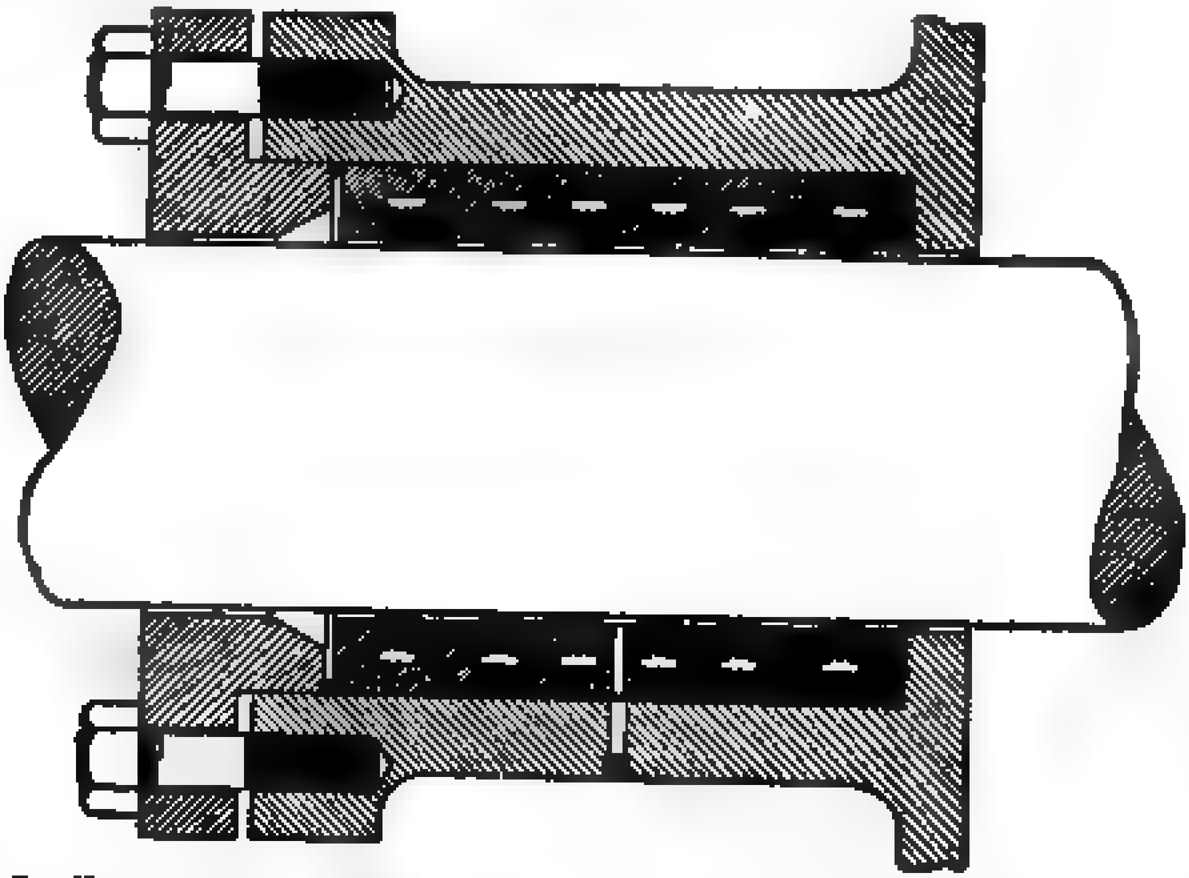


FIG. 1,082.—Stuffing box of the Lentz engine. The stuffing boxes are bored and ground to .001 in. of the exact dimension, and in them are placed a series of cast iron rings turned and ground to fit the stuffing box and accurately faced. These rings do not touch the piston rod. Interposed between them are five cast iron or "floating" rings having a square cross section which fits the piston rod very closely. Oil can be kept circulating in the stuffing box if necessary, but water (condensate) is sufficient to keep the box tight under considerable pressure.

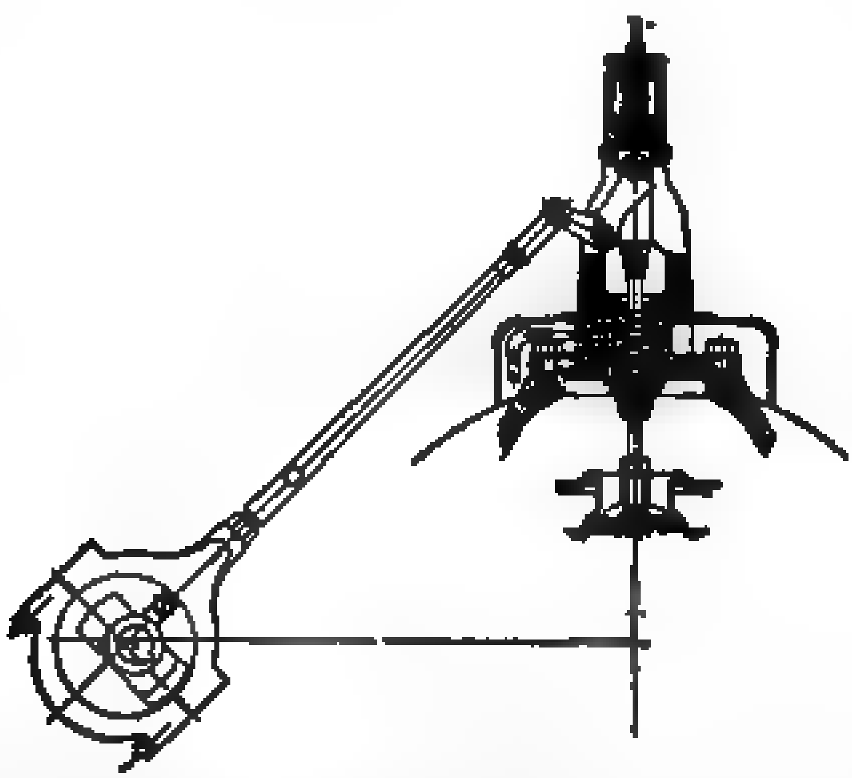


FIG. 1,083.—Sectional elevation of valve gear of Lentz engine. The valve is of the double seat poppet type, with seats not more than one-eighth inch wide. The valve stems which are ground to fit the long bushings shown have no stuffing boxes, the fit being to .001 inch. Grooves are turned in the spindles to prevent leakage. The valves are turned to such diameters that the lower one will just pass through the upper opening. No dash pots are used. The valve is moved by a cam acting on a roller. When the valve is seated the cam is not in contact with the roller, but the amount of clearance is too small to be seen in the cut. The roller is always in contact with the cam until the valve is seated.

Fig. 1,084.—Longitudinal section of the Lentz double … the valves and valve gear an interesting feature of this … used in the stuffing boxes. The framing is of the girder … the foundation bolts. The feet of the cylinder are free … nges of temperature. The piston body is turned slightly eccentric so as to get a large bearing surface on the bottom of the cylinder.

Fig. 1,085.—Transverse section of Lentz engine showing valve gear.

FIG. 1,083.—Governor as used on the Lentz engine.

port opening because there is no lap requiring linear advance. Poppet valves are sometimes called lift valves.*

There are various kinds of poppet valves, designed to meet the different conditions of operation, and they may be classed:

1. With respect to the number of ports, as

a. Single seated;
b. Double seated; etc.

2. With respect to construction, as

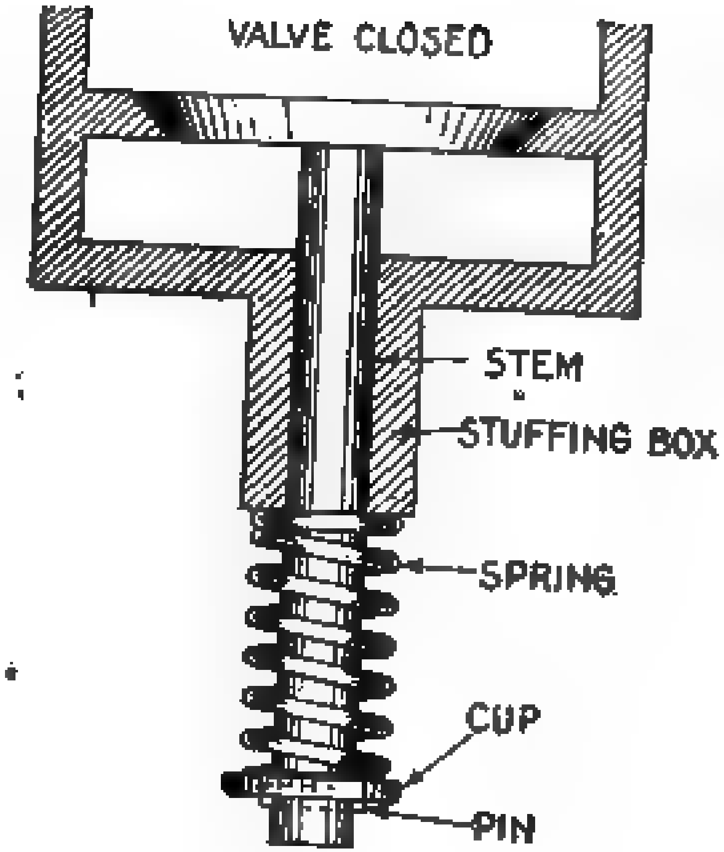
a. Solid; *b*. Hollow;
c. Bell shape.

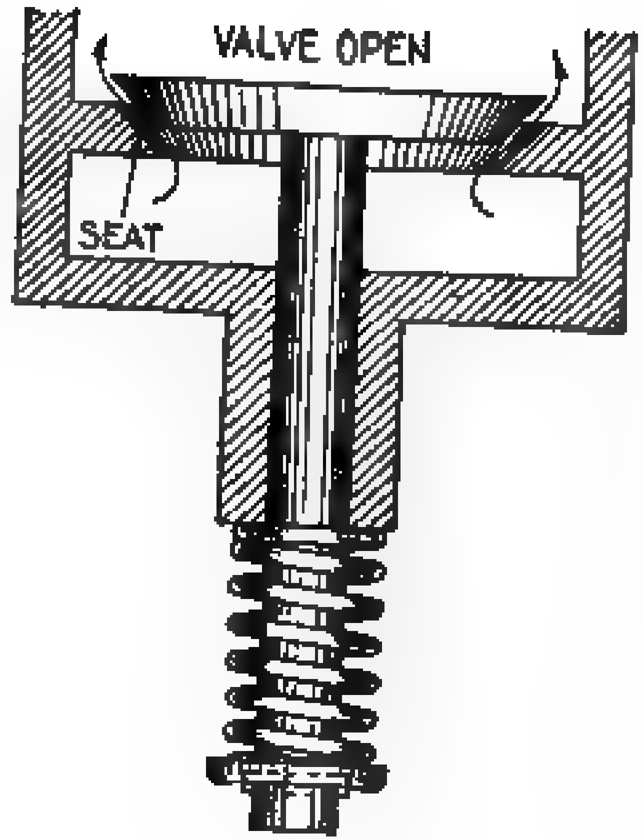
3. With respect to provision for prevention of leakage, as

a. Rigid; *b*. Expanding.

The main advantage of the poppet valve is that it does not move on its seat, thus requiring no lubrication. Hence, poppet valve engines are especially adapted to high pressure steam with considerable superheat. However, lubrication difficulties with sliding valves under high pressure superheated steam have been largely over estimated.

*NOTE.—It should be noted that the term lift applies to other types besides poppet valves; piston valves are sometimes arranged as lift valves.





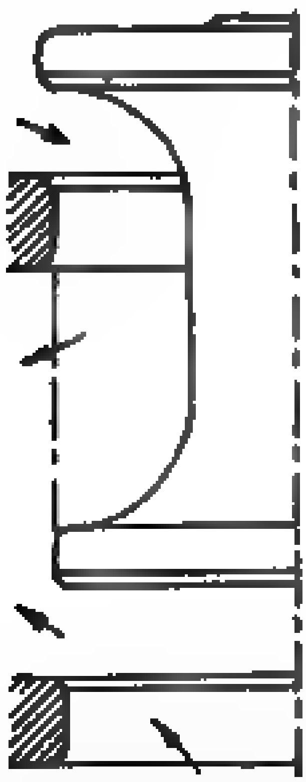
FIGS. 1,096 and 1,097.—Simplest form of poppet valve; fig. 1,096, valve closed; fig. 1,097, valve open. In this valve, which is the type generally used on gas engines, the valve gear operates at the lower end of the stem to open the valve, the latter being closed by the action of the spring.

F

section being called a *cage* or *basket*.

According to one manufacturer, the steam consumption of a full poppet valve engine is 16 lbs. per horse power hour or less, non-condensing with proper steam pressure and superheat. It should be noted that this high economy is due principally to the superheat, and not to any advantage inherent in the poppet valve beyond the fact that where variable cut off is obtained by the method of combined variable angular advance and variable throw a more adequate admission is obtained at very early cut off, because of the relatively small lift (travel) of the valve for full port opening as compared with sliding valves.

In comparing the economy of poppet valve engines working with high pressure highly superheated steam with other engines, especially those



FIGS. 1,101 to 1,103.—Typical poppet valve construction. Fig. 1,101 shows a solid double seat valve. While simple it requires a somewhat complicated valve chamber because of the top and bottom admission. Fig. 1,102 shows a hollow valve, a type used largely in Germany. The bell construction, fig. 1,103, is virtually the same as fig. 1,101 with the valve and seat interchanged. These valves and their cages are made of hand cast iron. The contact surface is inclined with respect to the valve stem the usual inclination being from 30° to 60° with the horizontal. The inclined seat does not cause so much change in the direction of flow of the steam and the speed of the valve perpendicular to the seat is not so great as the vertical speed, thus reducing valve hammer. Some authorities advise making the two cones of the two seats of such inclinations that they have a common apex on the assumption that under thermal changes the valve is deformed equally in all directions. In practice the width of the valve seat is small varying from one-eighth to one-half inch. The valves and seats after being machined, are ground to a steam tight fit.

having jacketed cylinders working with low pressure saturated steam, the factors of evaporation should be taken into account, and both reduced to the same basis, just as in the boiler tests the evaporation is reduced to "from and at 212° Fahr." value. When this is done it will be found that the apparent economy of high pressure superheated steam is relatively too high.

Figs. 1,096 and 1,097 show the simplest form of poppet valve in

closed and open position. This is a single seated valve and is the type usually employed on gas engines.

FIG. 1,104.—Valve gear side of Vilter poppet valve cylinder.

FIG. 1,105.—Crank side of Vilter poppet valve cylinder.

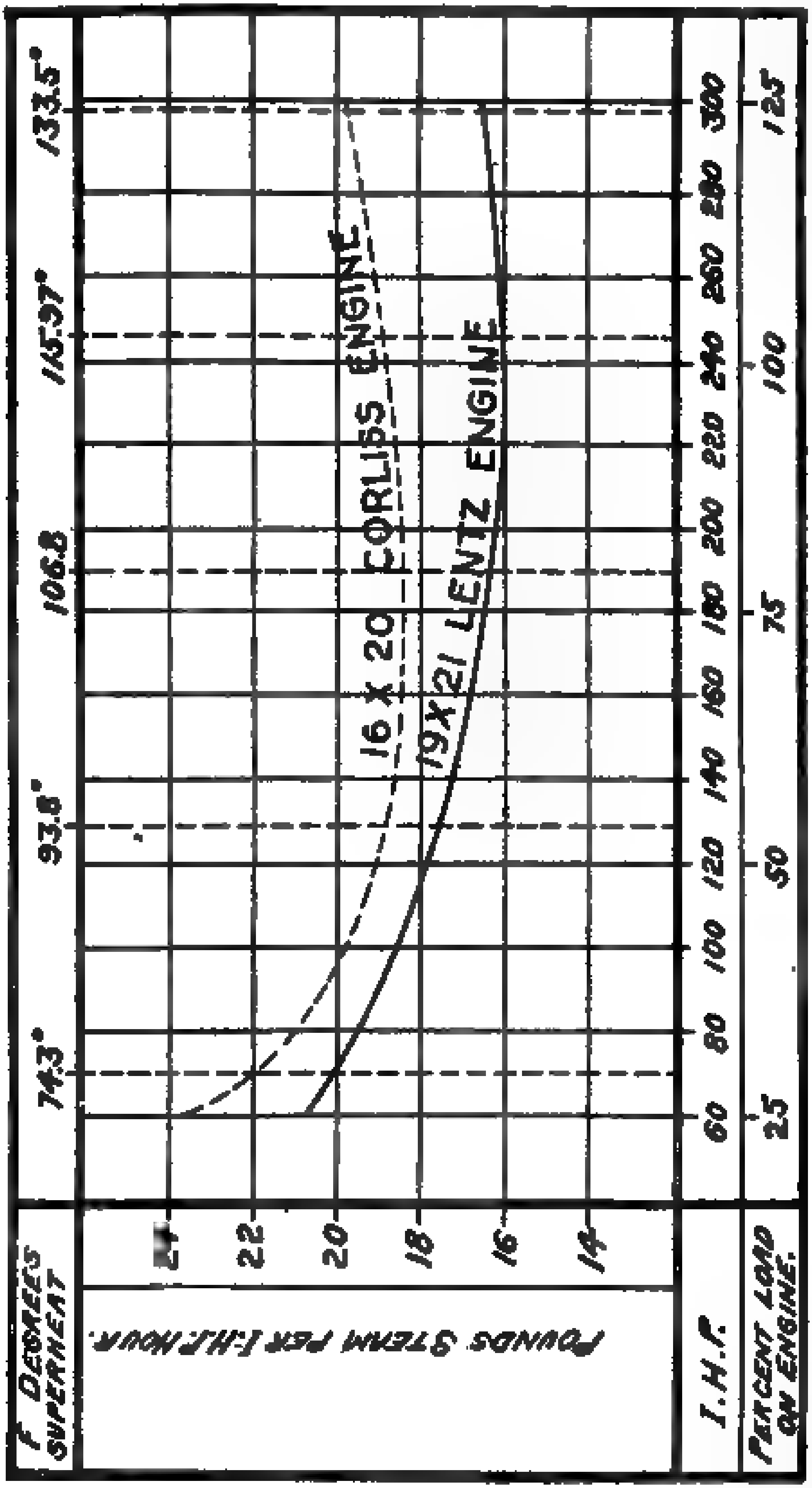


FIG. 1.1 … pressu… engine… curves… The … 200 r.p.m. and 135 lbs. … to favor the poppet valve … water cut off, hence the … allowing for these items.

Fig. 1,107.—Longitudinal section through Vilter poppet valve cylinder. One of the leading features of the engine is that the valve gear is operated through wrist plates driven from eccentrics placed on the engine shaft instead of employing a lay shaft. The steam and exhaust valves are of the double seated poppet type, mounted in cages inserted in suitable housings in the heads and cylinder. The valve stems are long and are ground into the guides. The stems are provided with labyrinth grooves which form a series of chambers in which any escaping steam is condensed, making them steam tight. The inlet valves are operated through rods from a wrist plate and by means of valve levers, the release mechanism being controlled by spring actuated, air cushioned dash pots provided with check valves. The range of cut off is from zero to .9. The ends of the lifting arms are provided with slide blocks which fit in corresponding sleeves in the valve stems in such a manner that any side pressure on the stems is avoided. The trip pins are adjustable for lap of the trip plates. The steam valves are controlled directly by means of a governor of the fly ball type designed to operate at several times the usual governor speed. A light single arm lever is connected from the governor to the governor crank on the front steam bonnet, the latter being again connected by another rod to a governor crank on the far steam bonnet. The steam valves are balanced. The exhaust valves are actuated through a separate eccentric and wrist plate, the valves being lifted by means of cams and rollers and returned to their seats by their own weight, assisted by springs. Both the steam and exhaust valve plates are arranged to be disconnected from the hook rods by means of a special disengaging device with which the rods may be released without danger of their falling to the floor or interfering with the other parts of the gear. When it is desired to release the rods, the catch pin is raised and given a quarter turn and the handle at the side is brought to its up position. The valve gear is then free to be moved by hand independently of the driving mechanism. By this means either the steam gear or exhaust gear may be disengaged and operated by hand for the purpose of turning the engine over, warming up the cylinder, etc., in the same manner as is done in the case of a Corliss engine. The action of the entire valve gear is analogous to that of the Corliss engine. The cylinder heads containing the admission valves are steam jacketed.

FIGS. 1,111 to 1,113.—Nordberg poppet exhaust valve, cage and bonnet. Fig. 1,111, assembly; fig. 1,112, valve and stem with labyrinth packing grooves in the latter; fig. 1,113, cage showing finished seat.

The valve and stem is in one piece, the forms being simply a solid disc with an inclined rim which is ground to fit the seat.

In operation, as shown in the figures, the valve opens by moving *away* from the seat instead of sliding on the seat as in the case of other types of valve. This, as already stated, is the chief advantage of poppet valves, thus requiring no lubrication which adapts them to operating at high temperatures as with superheated steam or in gas engines.

FIG. 1,114.—Sectional view of Nordberg poppet valve engine. *In construction*, the cylinder is a plain cylindrical casting similar to a piece of pipe, there being no ports or cored passages in the cylinder proper. The valves, valve cages, steam and exhaust ports and passages are all contained in the heads as shown. The steam enters from below by separate branch pipes to each cylinder head and sweeps up to the steam valves at the top, thus serving to steam jacket the ends of the cylinder. The exhaust outlets are in the center at the bottom of the heads, and where conditions permit, are connected by separate pipes beneath the engine, and behind the high pressure piping. For short stroke engines the exhaust nozzles on the heads are connected by a manifold designed to take up changes in length due to temperature; the exhaust is then piped from a single nozzle on this manifold.

Although only one sliding valve is necessary for engine operation, four are required when poppet valves are used.

The elementary construction shown in figs. 1,098 to 1,100 shows respectively two, three, and four ported valves, and figs. 1,101 to 1,103 typical construction.

According to one builder of poppet valve engines:

"Solid poppet valves, when ground for a certain steam pressure and

FIG. 1,117.—Cross section through valves of Nordburg poppet valve engine.

temperature, will remain steam tight indefinitely, providing the pressure or temperature is not allowed to vary; but if the pressure or temperature should vary—and it is next to impossible to prevent their changing—the difference in the coefficients of expansion of the two metals forming the valve and the seat in the cylinder will cause the one to expand more than the other, resulting in either the upper or the lower part of the valve not making contact with its seat in cylinder.

"To prevent the resultant leakage (which, of course, is never as great as with Corliss or piston valves, but is enough to make it an object to prevent), the self-expanding poppet valve, was designed.

"The upper part of this valve is not rigidly connected to the lower part, but allows an expanding and telescopic action and comes in contact with its seat shortly before the lower part with the result that both seats make steam tight contact irrespective of the difference in expansion of the cylinder and valve metal.

"With one grinding this valve has been steam tight with 159 pounds pressure and 150 degrees Fahrenheit superheat, and also with 100 pounds pressure, saturated steam.

"The telescopic action is never more than .003 or .004 inch, and steam tightness between the two elements of the valve itself is obtained by the peculiar construction of sprung metal packing rings."

It should be noted that the expansion principle as applied to poppet valves, introduces an extra joint and if this joint leak, the object sought will not be gained.

Poppet Corliss Engine.—This type of engine which consists of a poppet high pressure cylinder, and Corliss low pressure cylinder as shown in fig. 1,116, is especially suited to constant loads and high load factors; for example, electrolytic work and to those services such as driving pumps, air compressors, ammonia compressors, blowers, etc., where the mean effective pressure is constant and the engine may be designed for very high efficiency for practically a single set of conditions.

CHAPTER 18

UNI-FLOW ENGINES

This type of engine while comparatively new in this country, has been in use for many years in Europe and was invented by Prof. Johann Stumpf of Berlin, Germany, the object sought being *to reduce initial condensation.*

The principle of the uni-flow engine is that *steam enters at the ends of the cylinder and is exhausted through ports arranged around the center of the cylinder, and which are uncovered by the piston at the end of the stroke.* The steam has, consequently, a flow in but *one general direction* (from the ends toward the middle of the cylinder), hence the name uni-flow, or una-flow as so called by some builders.

In other types of engine, the steam returns on its path at the end of the stroke, and is exhausted at the same end of the cylinder at which it entered. By this method the relatively cold expanded steam "washes" the cylinder walls and heads during exhaust, thus absorbing more heat from them than it would if exhausted without a *return flow.* The uni-flow principle, as explained avoids this return flow with the result that the steam passages and heads have a higher temperature, thus initial condensation is reduced.

In the uni-flow engine, the steam enters at the end as in the ordinary engine, but is exhausted from the center of the cylinder at the furthest point from the heads. The flow of steam is in one direction only, hence the term uni-flow.

The advantage of this construction lies in the reduction of the loss by initial condensation. Ordinarily the cylinder head is chilled by the rush of exhaust steam past its face, so that at the beginning of the next stroke, a considerable per cent of the incoming steam is condensed as fast as it enters the cylinder.

In the uni-flow engine the hot ends of the cylinder are not chilled excessively in this manner and initial condensation is considerably reduced. Furthermore, unlike the four valve engine the face of the piston and cylinder head of the uni-flow engine are exposed to exhaust temperature only for the very small length of time that it takes the piston to uncover the exhaust ports.

On the return stroke the steam at exhaust temperature and pressure is caught between the piston and cylinder head, and as the piston moves back this steam is compressed; at the end of stroke, the pressure in the

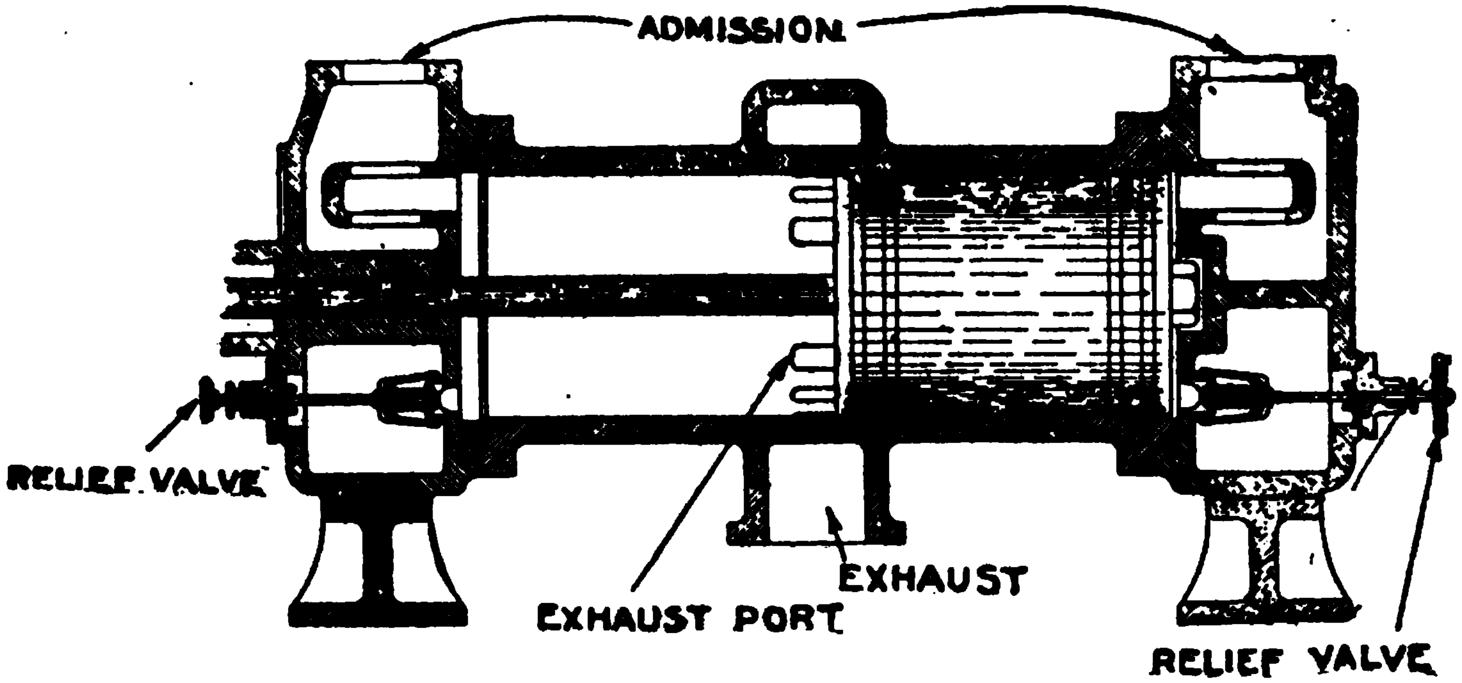
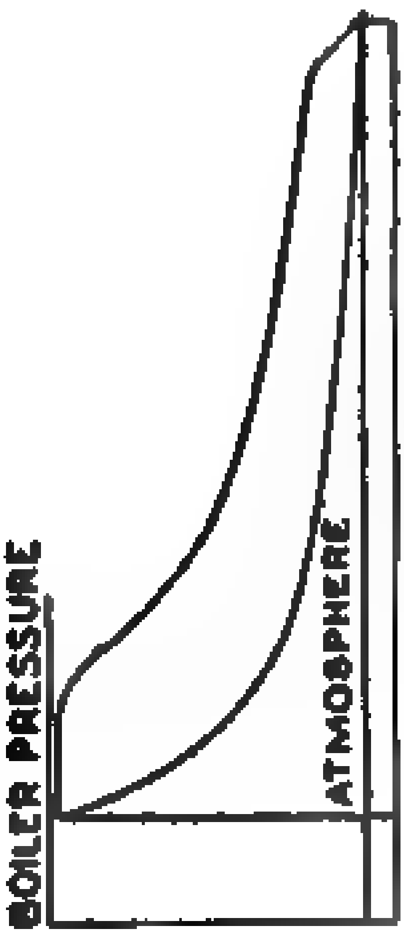


FIG. 1,118.—Typical European uni-flow cylinder of the ***relieved compression type.*** The cylinder is virtually two single acting cylinders, set end to end, with a common exhaust, the latter consisting of concentric ports at the middle as shown, these being uncovered by the piston when at the end of the stroke. ***In operation,*** steam is admitted by the admission valves at the ends of the cylinder. These valves (not shown) are usually of the double beat poppet type. After cut off the steam expands until near the end of the stroke the exhaust ports are uncovered by the piston and exhaust takes place, the piston is shown in the figure at the end of the stroke, exhaust ports fully opened. On the return stroke the piston closes the ports and compression begins very early. This early compression necessitates relief valves or equivalent to prevent excessive compression when operating non-condensing. Thus the uni-flow engine is inherently a condensing engine. The thermal disadvantage of the alternating flow engine consists mainly in the loss of heat through the cooling of large surfaces of the clearance space by the wet exhaust steam, which flows back with great velocity toward the exhaust valves, cooling the surfaces and carrying heat to the exhaust. This heat must be replaced by incoming steam, and considerable condensation is the result. In the uni-flow cylinder, this condensation is greatly reduced, as the heads are kept hot by the admission of steam at high pressure and by the heat of compression, while the exhaust ports are never exposed to the hot steam, so there can be no transfer of heat to be carried to waste. Thus a hot end and a cold end are maintained with a gradual fall of temperature between.

clearance space is equal to the steam line pressure. The temperature also increases not only due to the heat of compression but also to the absorption of heat from the head jacket, so that at the end of the stroke the steam temperature in the clearance spaces is considerably above the steam line temperature. Therefore, when the steam valve opens, the incoming steam meets relatively hot surfaces and condensation is largely reduced. The result is that a wider pressure and temperature range can be allowed in a single cylinder.



In Europe practically every steam plant operates condensing, but in America where there is an utter disregard for the preservation of the natural resources of the country, and where at present the cost of fuel is not so great, a large percentage of plants are run non-condensing.

As may be seen from the indicator card, fig. 1,119, which is typical of European uni-flow practice, compression begins as soon as the piston covers the central exhaust ports on its return stroke.

With this arrangement the compression period is about 90 per cent of the stroke, hence, it is evident that unless the pressure during exhaust be very low as when operating condensing, the lengthy compression would become so excessive as to be detrimental to the engine, unless large clearances were employed.

In this connection it should be noted that the same excessive compression will obtain if the vacuum fail on a condensing engine.

FIG. 1,121.—Indicator diagram for Filer and Stowell uni-flow engine. Cylinder 16×30; 125 lbs. initial pressure; 25 inch vacuum; 150 *r.p.m.* Economy given by manufacturers' test: 13⅓ lbs. saturated steam per horse power hour.

FIGS. 1,122 and 1,123.—Filer and Stowell variable clearance uni-flow engine details. Fig. 1,122, section of cylinder through exhaust; fig. 1,123 section of cylinder through steam valve and ports. The steam valve is of the piston type as shown.

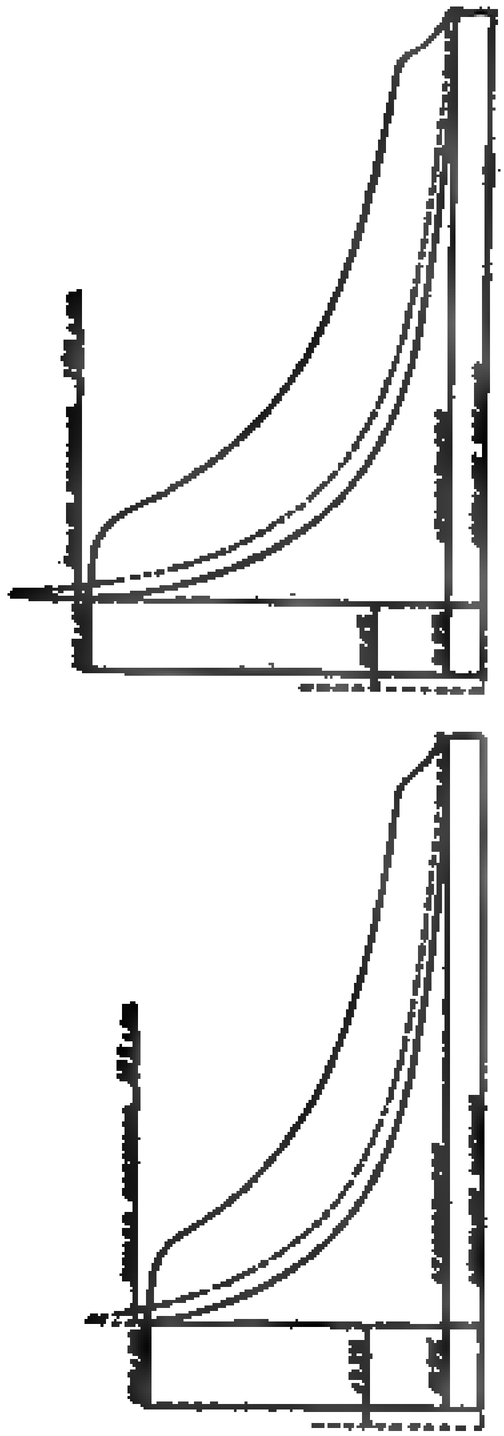


Fig.

in case the vacuum fail.

In order to meet this condition and to adapt the engine to non-condensing operation the uni-flow engine has been modified in several ways, giving rise to types which may be classed as:

1. Relieved compression;
2. Variable clearance;
3. Variable compression.

The first type employs snifting or relief valves which allow the compression to reach a predetermined degree only.

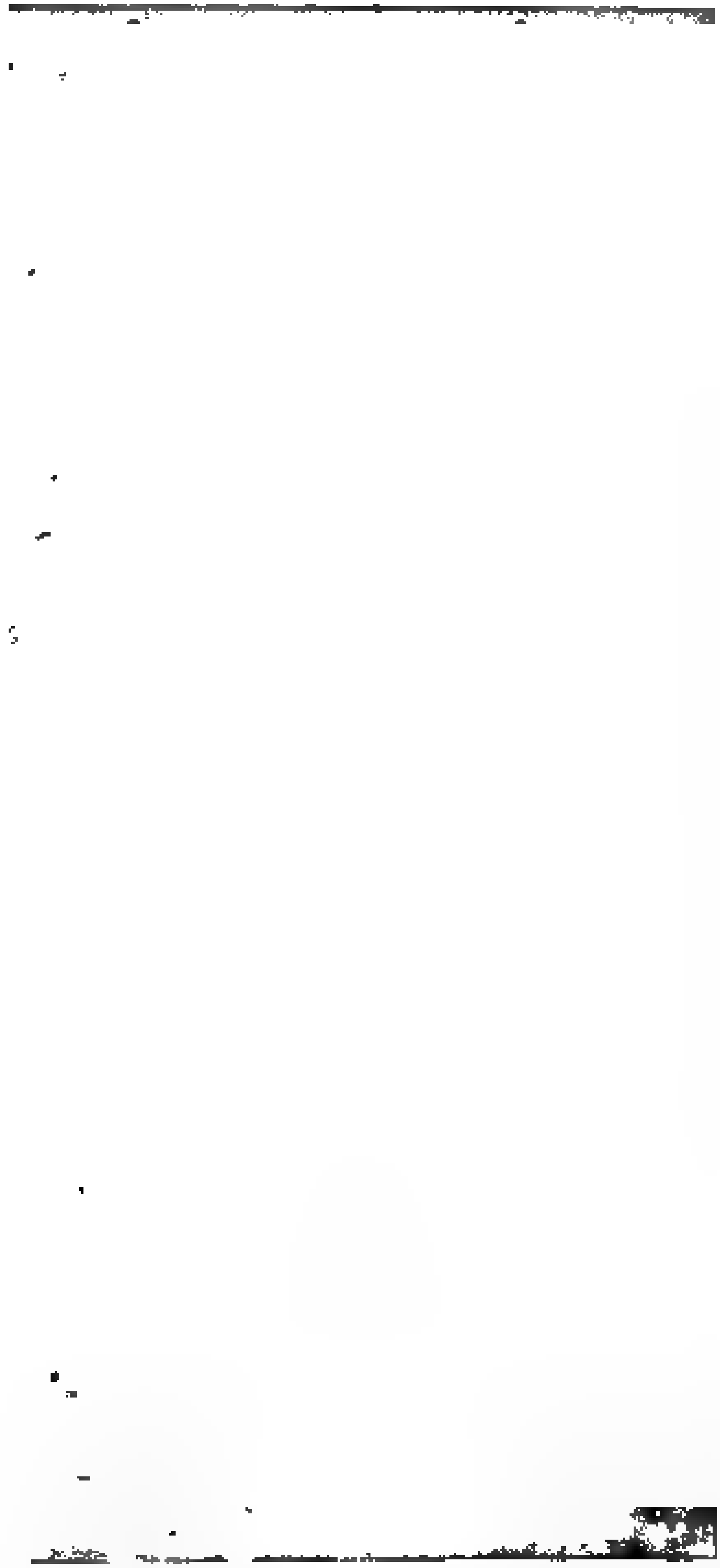
In the second type auxiliary clearance pockets are provided in both cylinder heads, controlled by hand operated or spring backed relief valves, which relieve the cylinder of excessive compression by forcing some of the compressed steam into the auxiliary clearance pockets.

FIG. 1,127.—Sectional view of Ames-Stumpf variable clearance uni-flow engine. The clearance is varied by an automatic by pass valve leading to auxiliary clearance pockets; the valve acts in case the vacuum fail, and thus prevents excessive compression. The engine has a Robb-Armstrong-Sweet governor connected to a pivoted eccentric, controlling the cut off by ***combined** variable travel and variable angular advance.*

FIG. 1,128.—Section through automatic by pass valve of Ames-Stumpf variable clearance uni-flow engine. The construction is simple, as shown, and the automatic operation of the valve is entirely controlled by existing conditions in the exhaust belt of the engine, these conditions being communicated from the exhaust belt of the engine to the valve piston through the small pipe connection between the exhaust belt and the piston chamber.

FIG. 1,129.—Skinner delayed compression uni-flow cylinder. The auxiliary exhaust ports, which are placed at the point in the cylinder where it is usual to start compression in an ordinary non-condensing engine, hence for non-condensing operation with this arrangement compression may be delayed, under control of the auxiliary exhaust valve, to a point which will avoid excessive compression.

FIGS. 1,130 and 1,131.—Plan and elevation of auxiliary exhaust valve gear of Skinner variable compression uni-flow engine. *In construction*, A, is the shaft supporting idler B, which is operated by cam C. This cam is operated by the engine valve gear which is connected to shaft D, on the outside of the cam box. When the cam C, raises the idler B, the latter raises the single beat exhaust valve, the stem of which projects within a short distance of the idler B. The spring around the valve stem has only enough tension to insure quick closing when operating at high speeds. Both cam and idler are immersed in oil as shown. The pocket E, is connected to the central exhaust port by means of a small pipe; the spring in the pocket is to keep the idler, through its attached shaft, in register with the stem of valve and cam C, directly underneath. *In operation*, when the vacuum reaches a pre-determined point, it overcomes the tension of the spring in pocket E, and draws the shaft into the pocket, this in turn shifts the idler to position B′, out of register with cam C, so that while the cam still operates as before, it cannot lift the valve. Hence the valve remains closed at all times when vacuum exists in the exhaust pipe, and is kept tightly closed by the pressure in the cylinder. Bridge F, acts as a slide and a rest for the idler, and holds it at the proper height so that if the vacuum fail, the spring will slide the idler between the cam and valve stem, placing auxiliary exhaust valve in gear for non-condensing operation.



FIGS. 1,132 to 1,137.—Sectional views of Skinner delayed compression uni-flow cylinder and indicator diagram illustrating non-condensing cycle. ***In operation,*** fig 1,132 ***head end,*** admission taking place on dead center; *crank end*, exhaust taking place through main ports only; fig. 1 133 *head end*, full opening of valve; *crank end*, exhaust through main ports about to close, fig 1,134, *head end*, point of cut off, *crank end*, exhaust of trapped steam through open auxiliary exhaust valve; fig. 1,135, *head end*, expansion taking place, *crank end*, beginning of "delayed" compression by piston covering auxiliary exhaust ports; fig. 1,136, *head end*, pre-release begins, *crank end*, continuation of compression, fig. 1,137, indicator diagram illustrating the cycle of operation just described.

Since the added clearance results in less economy, this type is only suitable for condensing operation; the auxiliary clearance being intended

FIGS. 1,144 to 1,147.—Cross sections of Skinner expanding poppet valve and parts unassembled. *In construction* the upper part of the valve is not rigidly connected to the lower part, but allows an expanding and telescopic action to compensate for any difference in expansion of the cylinder and valve metal. The spring shown is not essential to the operation of the valve, its function being to serve as an elastic stop.

FIGS. 1,148 and 1,149.—Views of Skinner auxiliary valve. Fig. 1,148, valve closed and with cover off showing mechanism; fig. 1,149 valve open. Labyrinth or water groove packing is used.

for use in starting the engine without a vacuum, or for emergency should the vacuum fail.

The third type employs, besides the main exhaust ports at the middle of the cylinder, auxiliary exhaust valves located

FIG. 1,150.—Detail of admission gear of Skinner variable compression uni-flow engine. *It consists of* one rocker arm, with removable cams, which dip into an oil bath and control the operation of the valves by means of forked levers or lifters, on which are mounted rollers bearing against the cams. The valve spring is telescoped in the housing (not in the steam space). Labyrinth or water groove packing is used, as here shown and also in fig. 1,148. The valve, cam, and lifter are adjusted by a set screw located above the lifter.

about midway between the ends and the middle of the cylinder, thus, by exhausting through these auxiliary exhaust valves,

FIGS. 1,151 and 1,152.—Longitudinal and cross sections of Nordberg relieved compression poppet uni-flow engine, showing arrangement of valves and valve gear.

after the piston has closed the main exhaust ports, compression may be delayed to such point that the pressure at the end of compression will not be excessive.

The various types are illustrated in the accompanying cuts.

NOTE.—Clearances in uni-flow engines are 1½ to 2%. The mechanical efficiency (simple) is .88 to .89. Some tests are as follows: ***Test 1:*** observers, Burmeister and Wain; engine 184 h. p.; steam sat. 140 lbs.; 13.64 lbs. steam per h. p. hour. ***Test 2:*** observer, Stumph; engine 24½ × 39; steam 140 lbs., 376° Fahr.; 12.25 lbs. steam per h. p. hour. ***Test 3:*** observer, Stumpf; steam 120 lbs., 490° Fahr.; 10.85 lbs. steam per h. p. hour. ***Test 4:*** observer, Stumpf; steam 162 lbs., 518° Fahr.; 9.9 lbs. per h. p. hour. ***Test 5:*** observers, Burmeister and Wain; engine 115 to 220 h. p.; steam 140 lbs., 667° Fahr.; 9.06 to 9.68 lbs. steam per h. p. hour. The above engines were unjacketed single cylinder.

NOTE.—Tests on 100 h. p. condensing engine by Lentz gave the following results:

Initial steam pressure

235	461

Steam temperature Fahr.

923°	1,018°

Lbs. steam per i. h. p. hour

6.52	5.67

B. t. u. per i. h. p. min.

162	144

CHAPTER 19

COMPOUND, TRIPLE AND QUADRUPLE EXPANSION ENGINES

To secure economy in the use of steam in the operation of a steam engine, it is necessary that the steam be expanded to a proper degree, depending on:

1. Initial pressure, and
2. Back pressure.

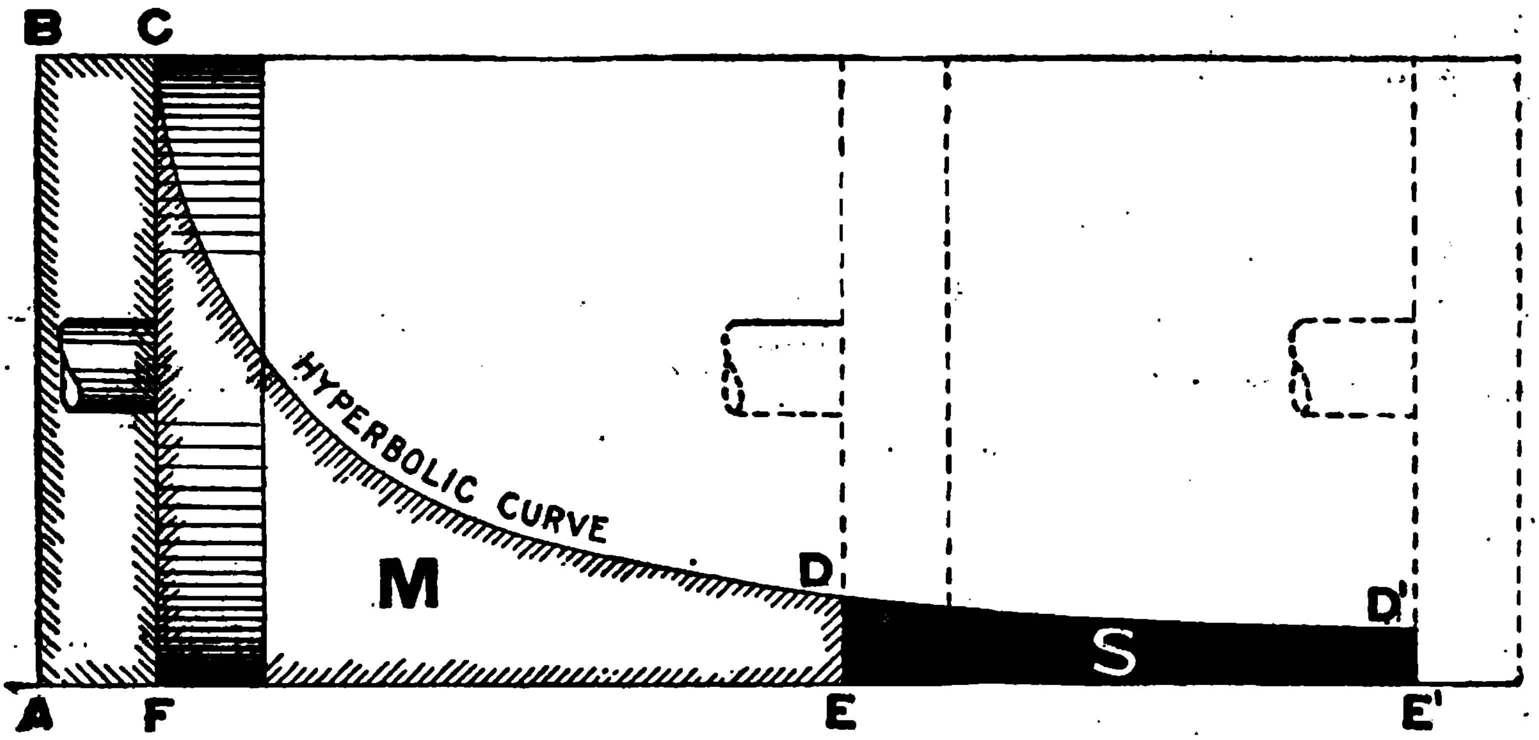


FIG. 1,153.—Diagram illustrating the theoretical economy due to expansion.

Theoretically, the greater the number of expansions, the greater the economy, as is clearly illustrated in fig. 1,153.

Thus, in the figure, if an engine have a stroke corresponding to A F, and steam be admitted full stroke to F, as would be the case in some steam pumps, considerable steam would be required in proportion to the work done, because it was not allowed to expand. In this case the work done during the stroke would be represented by the area A B C F.

Now suppose the stroke be increased to E, and the same volume of steam

(represented by A B C F) be admitted; it will expand from C to D, and the gain due to expansion will be equal to the area C D E F or **M**

Similarly, if the stroke be increased to E′ and the same amount of steam be admitted there will be an additional gain equal to the black area D D′ E′ E or **S**, the total gain being **M+S.**

It would appear from the foregoing that the greater the expansion, the greater the gain, *but this does not hold in practice.* There are several conditions which limit the number of expansions that can be made *in one cylinder* without introducing losses

FIG. 1,154.—Diagram illustrating the loss due to condensation.

that would offset the gain. These losses which increase with the number of expansions are due to:

1. Initial condensation;
2. Condensation during expansion;
3. Clearance;
4. Decreasing mechanical efficiency;
5. Interest in the increased cost of engine.

Condensation.—The principal loss is by condensation, the effect of which is to reduce the area of the card; thus, instead

of obtaining the theoretical card, as in fig. 1,153, it would take a form similar to that shown in fig. 1,154.

In the figure, assume the initial pressure to be 120 lbs. absolute corresponding to 341° Fahr., terminal pressure at D′, 10 lbs. and back pressure 2 lbs. corresponding to 126°. Then if the expansions take place in a single cylinder, the range of temperature in that cylinder is 341° — 126° = 215°, and in operation, the cylinder walls being subjected alternately to the high and low temperatures, will be heated to some intermediate temperature. Accordingly, when steam is admitted from B, to C, at the high temperature corresponding to 120 lbs., it meets the comparatively cold walls and considerable *initial condensation* takes place, as indicated by the shaded area X representing a pool of water at the bottom of the cylinder.

The temperature of the steam expanding from C, decreases as its volume increases and condensation continues until the temperature of the steam is reduced to that of the cylinder walls, the two temperatures being equal at some point of the stroke as M, the condensation which has taken place during this period being represented by the shaded area Y. As the piston moves beyond M, the temperature of the steam becomes *lower* than that of the walls and *re-evaporation* takes place, that is, part of the water in the bottom of the cylinder flashes into steam. Thus, if *p n*, represent the height of the water in the bottom of the cylinder when the piston is at M, part of the water will be changed into steam as the piston moves to the end of the stroke and the amount of water in the cylinder at the end of the stroke will be less than when the piston was at M, as indicated by the sloping line *n t*. The effect of this re-evaporation is to increase the area of the card by the solid black area R, thus elevating the expansion line so that it runs from M, to O′, instead of to O. This gain indicated by the solid black area R, is, as can be seen, small, compared to the condensation loss indicated by the shaded area G. Moreover, it must be evident, that any water in the cylinder as E *t*, not re-evaporated before the end of the stroke is lost during exhaust.

It must be evident also that if more than one cylinder be provided so that the expansion can take place in two or more stages, thus reducing the temperature range in any one cylinder, it will result in less loss by condensation and the curve CMO′, fig. 1,154, will approach the saturation curve, thus increasing the efficiency.

FIGS. 1,155 and 1,156.—Diagrams showing the effect on the clearance loss due to increasing the number of expansions in a single cylinder engine.

Effect of Clearance.—Clearance is usually expressed as a percentage of the cylinder volume, but so far as economy is concerned its relation to the cylinder volume up to the point of cut off should be considered as must be evident from figs. 1,155 and 1,156.

Thus, suppose a single cylinder variable cut off engine have a clearance of 5 per cent of the cylinder volume.

If this engine cut off at say, ½ stroke as in fig. 1,155, the steam required to fill the clearance (represented by the solid black area) will be 10 per cent of that supplied during admission represented by the shaded area M.

Now, if the cut off be reduced to one-tenth stroke as in fig. 1,156 then 50 per cent of the volume of steam admitted from the beginning of the stroke to the point of cut off is required to fill the clearance.

Of course, in practice, part of the steam required for clearance is obtained by compression, however, as the illustrations clearly indicate, the loss due to clearance increases as the cut off is shortened.

If the expansions take place in two or more cylinders, this loss will be considerably reduced.

Compound Engines.—A compound engine is one in which the steam is expanded in two stages; the steam after expanding in one cylinder is further expanded in a second cylinder. The first or small cylinder is called the *high pressure cylinder*, and the second or large cylinder, the *low pressure cylinder*.

Sometimes, the second stage of the expansion is effected in two cylinders, making the number of cylinders greater than the number of stages, but in general, such construction is questionable and open to criticism.*

The term compound was formerly, though improperly applied to triple and quadruple expansion engines.

Compound engines may be classified:

1. With respect to the action of the steam, as

   a. Direct expansion;
   b. Receiver { without drop. / with drop.

2. With respect to construction, as

   a. Tandem;
   b. Cross;
   c. Three cylinder;
   d. Steeple.

3. With respect to service, as

   a. Stationary;
   b. Marine;
   c. Locomotive.

The two types of special interest here are the direct expansion engine, and the receiver engine.

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*NOTE.—The author does not believe that three cylinder compounds, or four cylinder triple expansion engines are justified under any conditions, for as Watt has said, "*the supreme excellency of machinery is in its simplicity*" and the above types represent anything but simplicity, taking into account their working cycles. If three cylinders are to be provided, involving three sets of working parts, such complications seem not justified without gaining the economy due to three stage expansion, and similarly when four cylinders are used the engine should be a quadruple.

***Direct Expansion Compound Engine.**—Fig. 1,157 shows the essential features of the direct expansion engine.

As shown, steam enters at **A**, and flows to the high pressure piston **B**, which is just beginning its stroke, being cut off and expanded as in a simple engine.

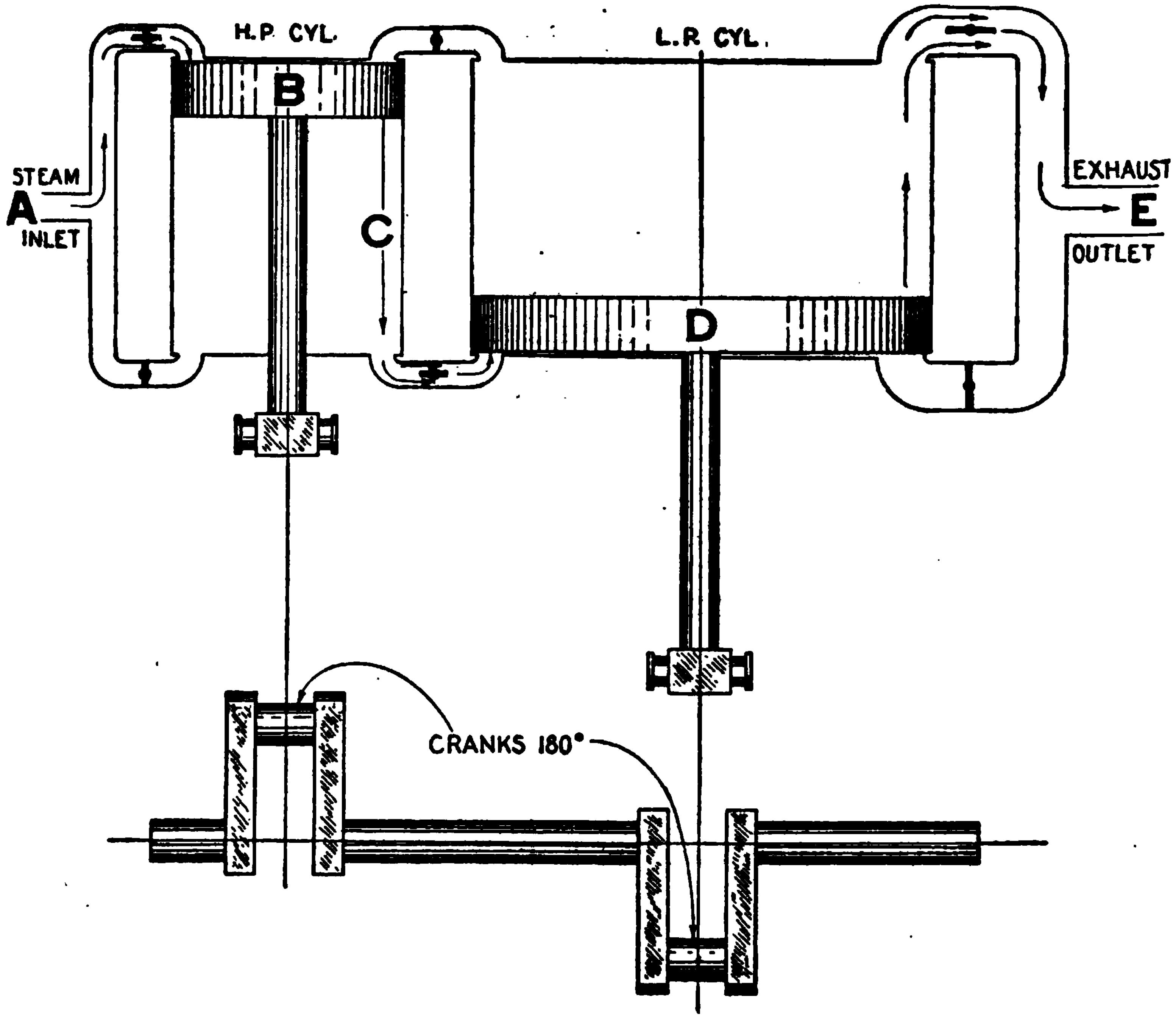

FIG. 1,157.—Elementary ***direct expansion compound engine.*** **The expansion is continuous after cut off in the** ***h.p.*** **cylinder, steam being admitted to the** ***l.p.*** **cylinder during the entire stroke.**

During the stroke, the steam **C**, admitted to the other side of the piston and expanded during the previous up stroke, is now exhausted from the high pressure cylinder by the decending piston directly into the low pressure cylinder against the piston which is moving upward.

***NOTE.—The term direct expansion compound engine is applied to an engine with cranks at 180° so that the *h.p.* cylinder exhausts directly into the *l.p.* cylinder instead of into a *receiver* as in an engine with cranks at 90°. Thus the expansion line on the combined diagram is practically *continuous* instead of being interrupted by *drop* in the receiver.**

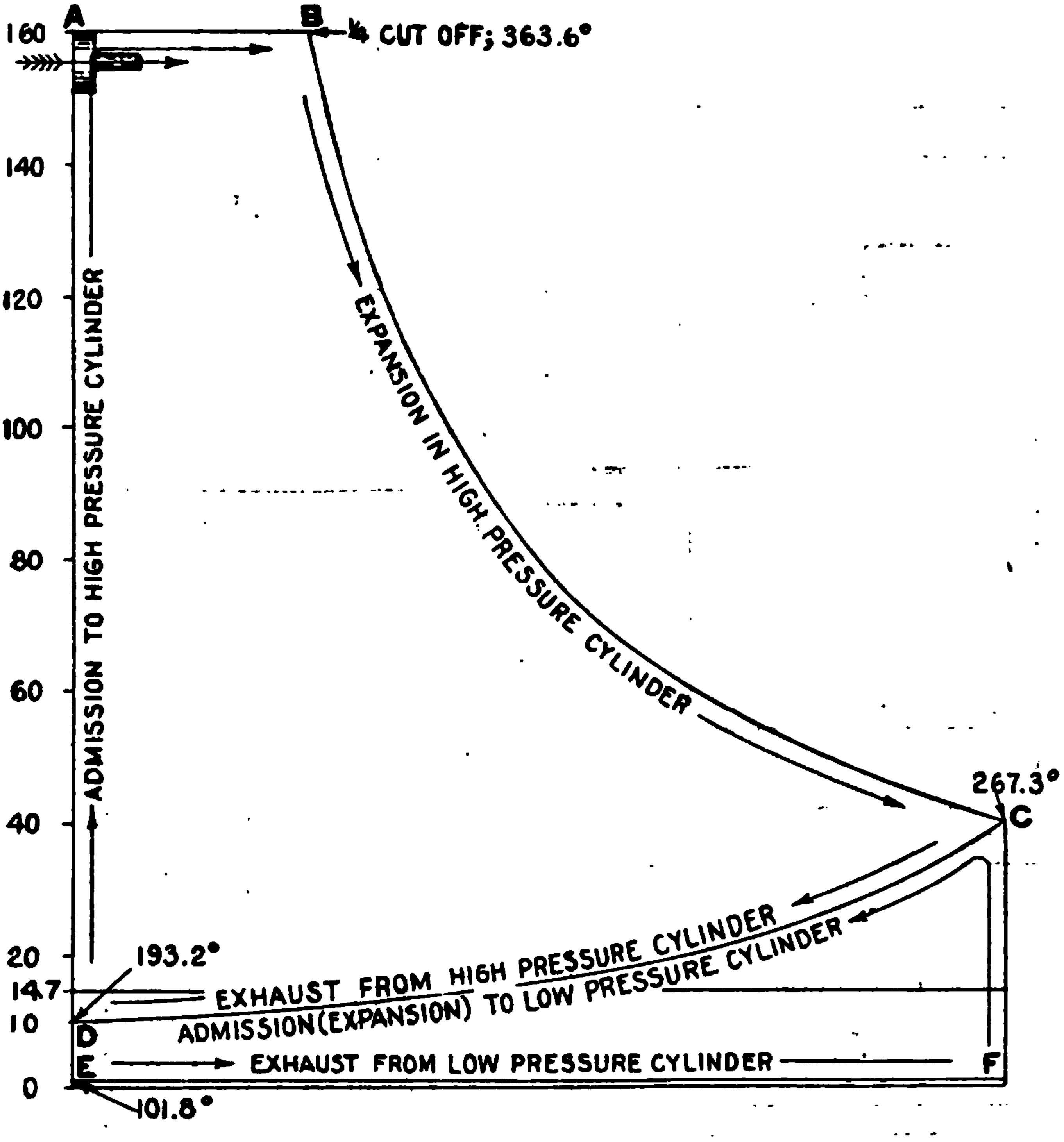

FIG. 1,158.—Theoretical diagram of ***direct expansion compound engine*** (fig. 1,157), illustrating its cycle of operation. ABCD, is the diagram for the *h.p.* cylinder, and CDEF, the diagram for the *l.p.* cylinder. ***As shown***, steam is admitted to the *h.p.* cylinder from D, to B, expanded from B, to C, and exhausted into the *l.p.* cylinder from C, to D, thus completing the cycle in the *h.p.* cylinder. It will be noted that the exhaust line of the *h.p.* cylinder forms the admission line of the *l.p.* cylinder. The cycle of the low pressure cylinder begins at F, but since clearance is assumed to be zero, the pressure immediately jumps to C, when the *h.p.* exhaust valve opens. There is no cut off in the *l.p.* cylinder; hence the expansion is continuous from the point of cut off in the *h.p.* cylinder to release in the *l.p.* cylinder. The steam is released at D, and the pressure immediately falls to the exhaust pressure which is here taken at 1 lb. corresponding to a 27.88 in. vacuum. As shown, the value shere taken are: initial pressure 160 lbs.; terminal pressure 10 lbs.; back pressure 1 lb.; cut off one-fourth stroke. The total number of expansion then is $160 \div 10 = 16$, and the terminal pressure at C, in the *h.p.* cylinder is $160 \div 4 = 40$ lbs. The total range of temperature is $363.6° - 101.8° = 261.8°$, of which range in the *h.p.* cylinder is $363.6° - 193.2° = 170.4°$ and in the *l.p.* cylinder is $267.3° - 101.8° = 165.5°$. It will be noted that while the two ranges are about the same, which is desirable, the temperatures overlap, making the sum of the two ranges $170.4° + 165.5° = 335.9°$ or 74.1° in excess of the total range of temperature, that is, the *l.p.* cylinder walls are subjected to 74.1° excess cooling during exhaust which is an inherent defect of the cycle.

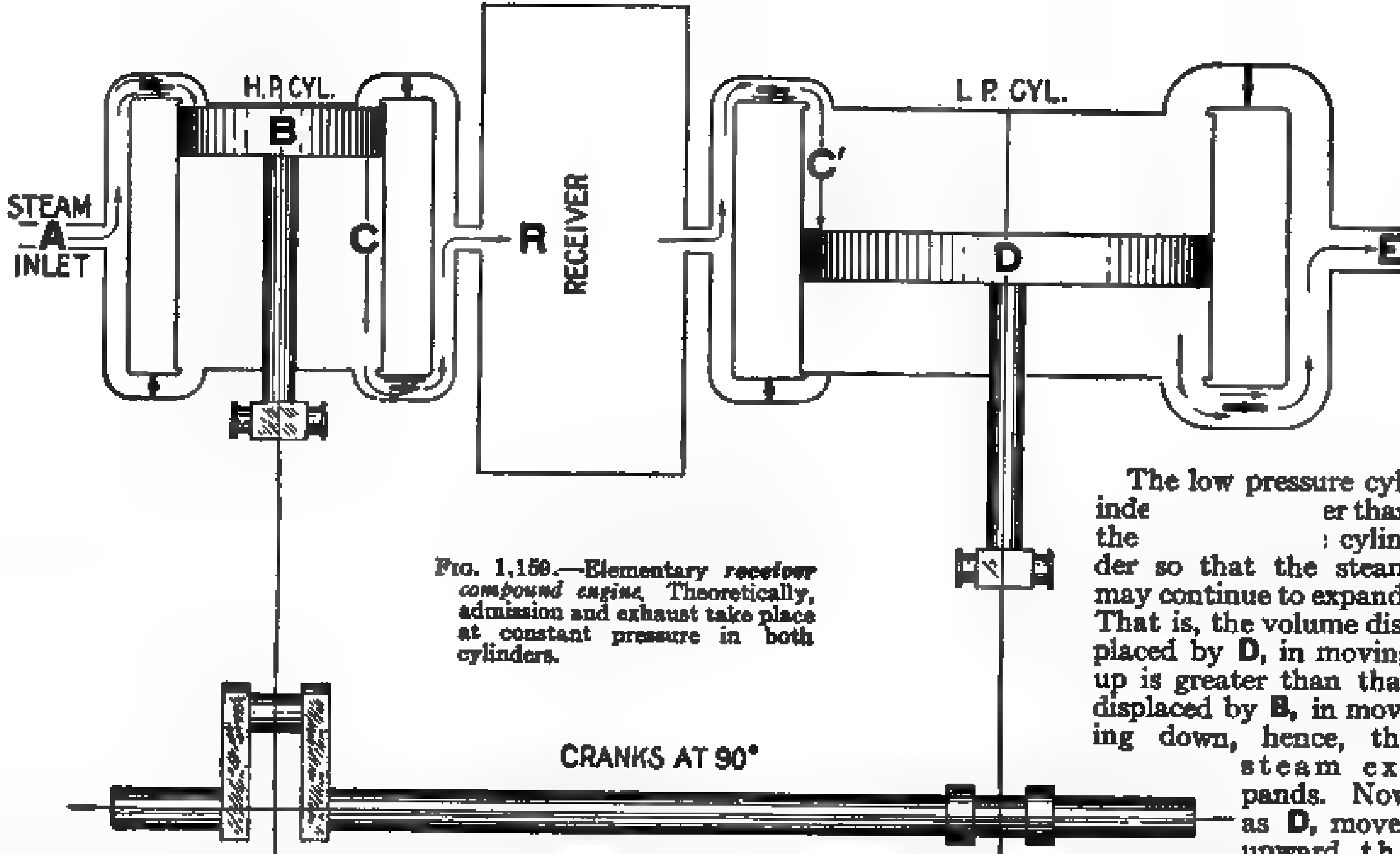

Fig. 1,159.—Elementary *receiver compound engine.* Theoretically, admission and exhaust take place at constant pressure in both cylinders.

The low pressure cylinde er than the cylinder so that the steam may continue to expand. That is, the volume displaced by **D**, in moving up is greater than that displaced by **B**, in moving down, hence, the steam expands. Now as **D**, moves upward, the steam admitted to the upper side during the previous stroke is exhausted to the atmosphere or condenser at **E**, according as the engine is run non-condensing or condensing. The cycle of operation just described is illustrated in the theoretical diagram, fig. 1,158.

Receiver Compound Engine.—In this type, fig. 1,159, the exhaust from the high

pressure cylinder passes into a receiver instead of direct to the low pressure cylinder.

As shown, steam enters at A and flows to the *h. p.* piston B, which is just beginning its stroke, being cut off and expanded as in a simple engine.

During the stroke the steam C, admitted to the other side of the piston and expanded during the previous up stroke, is now exhausted from the *h. p.* cylinder by the descending piston into the receiver at R, as indicated by the arrows. The *l. p.* crank being 90° ahead of the *l. p.* piston has

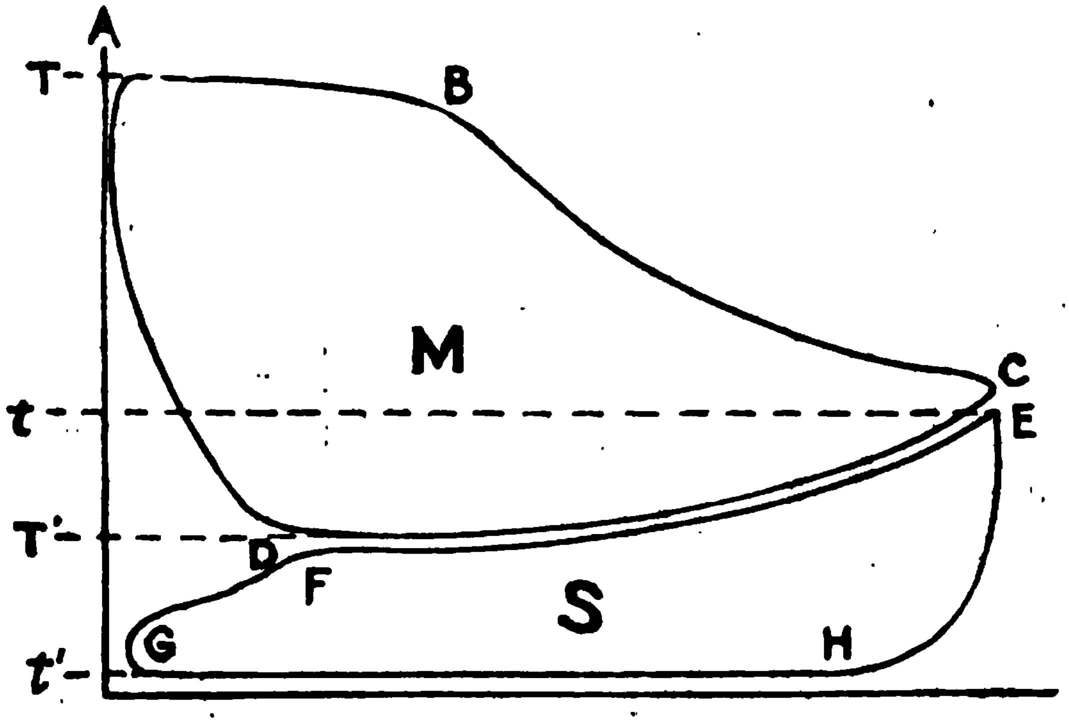

FIG. 1,160.—Actual diagrams of a single acting compound engine with cranks at 180° showing effect of very small receiver. M, is the *h. p.* diagram and S, the *l. p.* diagram. In the *h. p.* cylinder, admission is from A, to B, and expansion from B, to C. At C, release occurs and steam exhausts into the *l. p.* cylinder from C, to D, when the exhaust valve closes and compression begins. The exhaust line C D of the *h. p.* cylinder is an expansion curve because the steam expands from one cylinder into the other. The slight gap between C D and E F, is due to *free expansion* and friction in the ports and passages. E D, is the admission line of *l. p.* cylinder. At D, communication between the two cylinders is shut off, the steam in the *h. p.* cylinder being compressed to A, and that in the *l. p.* cylinder expanded from F, to G. Exhaust taking place from G, to H, and compression from H, to E. The temperature ranges are TT′, for the *h. p.* cylinder, and *t t′* for the *l. p.* cylinder, the ranges overlapping from *t* to T′.

moved about one-half stroke, and assuming an early cut off in the *l. p.* cylinder, steam has been admitted to the *l. p.* cylinder, and cut off as indicated by the broken arrow C′, expansion now taking place in the *l. p.* cylinder and simultaneously exhausting from the lower side of the piston D, into the atmosphere or condenser at E, according as the engine is run non-condensing or condensing.

Theoretically, as must be evident, if steam be cut off in the low pressure cylinder at such point that the volume admitted to the *l. p.* cylinder is the same as that exhausted from the *h. p.* cylinder, and the receiver be

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taken as in fig. 1,162. As shown the temperature range in the *h.p.* cyl. is 363.6° − 267.3° = 96.3°, and in the *l.p.* cyl., 267.3° − 101.8° = 165.5°. The ranges do not overlap as in fig. 1,159, but are not so near equally divided, showing a difference of 165.5° − 96.3° = 69.2° excess in the *l.p.* cyl. Now, if steam be cut off at one-fourth stroke in the *h.p.* cyl. and DC, represent the volume of the *h.p.* cyl., then GF, will represent the volume of the *l.p.* cyl. and DC + GF, the cut off in the *l.p.* cyl. to maintain a constant pressure in the receiver and give the continuous expansion line BCE.

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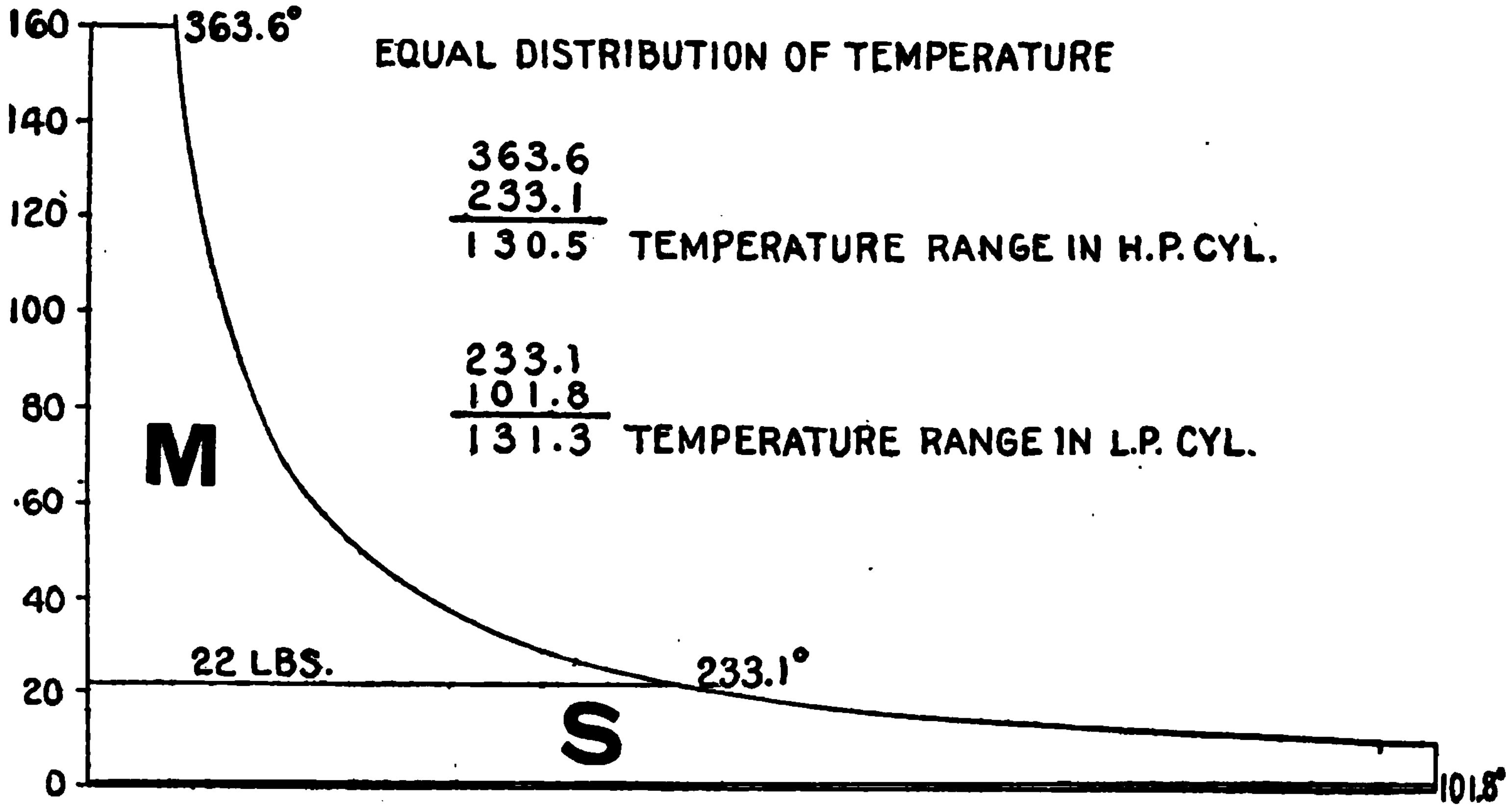
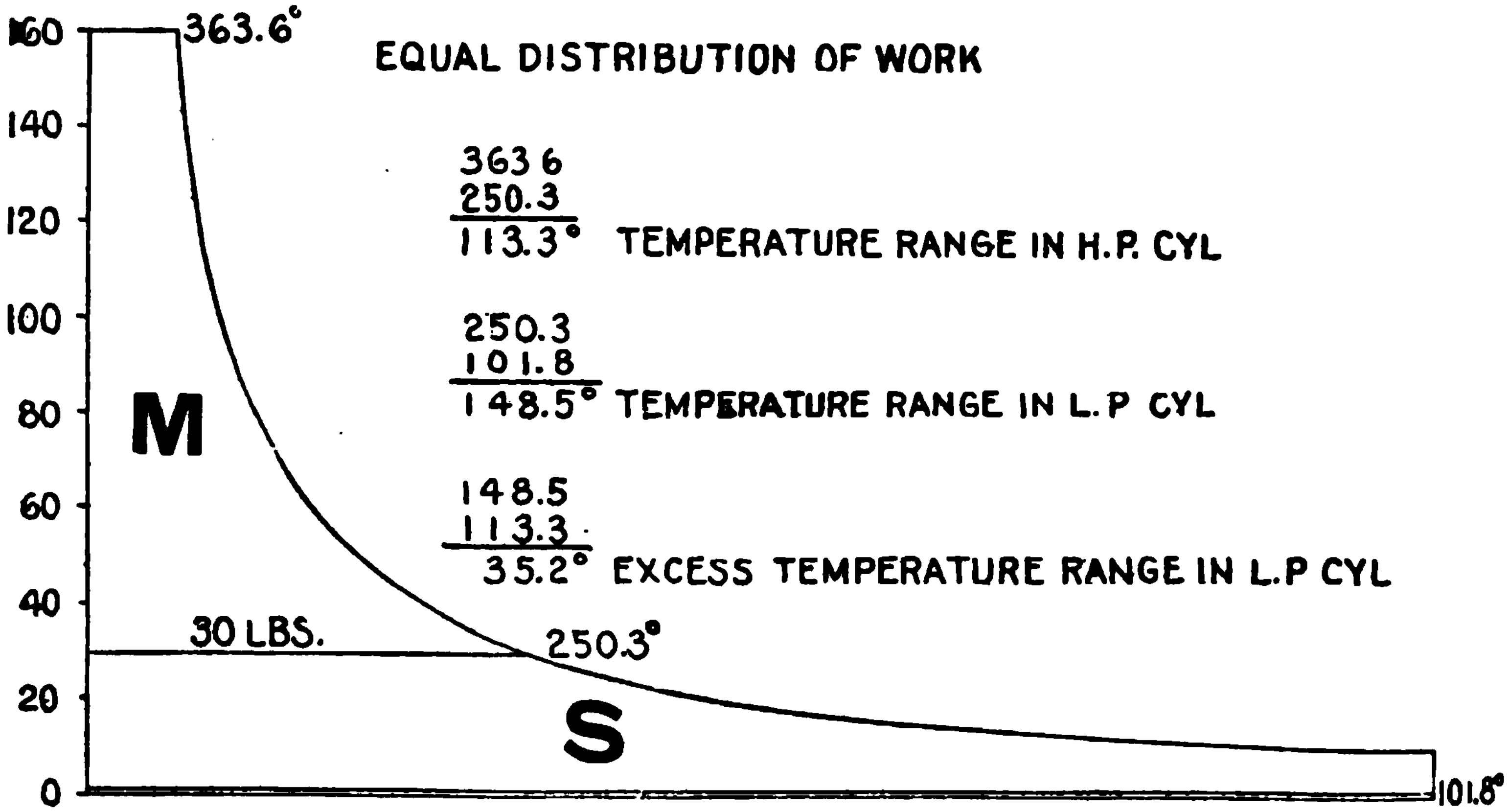
at C, the pressure will *drop* to the receiver pressure *c*, which depends on the cut off in the *l.p.* cyl. Hence, exhaust from the *h.p.* cyl. on the return stroke will be represented by the line *cd*, instead of CD. *d*E, represents admission to the *l.p.* cyl., EF, expansion; FG, release and GH, exhaust The two areas M and S, represent work done in the *h.p.* and *l.p.* cyl. respectively, and the solid black triangular area CE*c*, the loss due to drop by free expansion in the receiver.





FIGS. 1,163 and 1,164.—Combined theoretical diagrams for compound engine; fig. 1,163 diagrams for equal temperature range in each cylinder; fig. 1,164 diagrams for equal work in each cylinder. For equal ranges of temperature, fig. 1,163, the *h.p.* cyl. will develop the greater power, whereas, when both cylinders develop the same power as in fig. 1,164 the *l.p.* cyl. will be subjected to a larger range of temperature, being in this case 148.5° − 113.3° = 35.2° in excess of that in the *h.p.* cyl. In this connection it should be noted that for low pressure as in the *l.p.* cyl. the difference in temperature per lb. difference in steam pressure is much greater than for high pressures as in the *h.p.* cyl. Thus the difference in temperature of steam at 160 and 159 lbs. pressure is .5° while for 10 and 9 lbs. steam pressures the difference is 4.9°.

In marine engines, which employ a late cut off, this loss is considerable, as will be later shown.

Cylinder Ratio in Compound Engines.—It must be evident that if steam is to expand in both cylinders of a compound engine, the second or low pressure cylinder must be larger than the first or high pressure cylinder, for as stated by Clark:

"If the capacity of the cylinders were equal, it is clear that the first cylinder full of steam would only be transferred to another space of the same volume. No useful work would be done by this operation, for though the translated steam would press forwards on the second piston, according to its elasticity, it would likewise press backwards with equal force on the first piston, and so counteract itself. To generate useful work by expansion, therefore, in exhausting steam from the first into the second cylinder, the second cylinder must be of greater capacity than the first."

The relative size of the low pressure cylinder as compared with the high pressure cylinder, that is, the *cylinder ratio* is of considerable importance, as, upon this depends the economical working of the engine.

The choice of cylinder ratio is governed by a number of conditions, more or less conflicting, such as

1. Initial pressure;
2. Quality of the steam;
3. Terminal pressure;
4. Character of the load;
5. Kind of service, etc.

The items upon which the cylinder ratio directly depends, are:

1. Initial pressure;
2. Terminal pressure;
3. Cut off;
4. Drop.

The initial pressure will be governed by the degree of economy desired which in turn is influenced by the cost of fuel, and whether the engine is to be run non-condensing or condensing.

The terminal pressure is also governed by the degree of economy desired, and limited by the character of the load and the back pressure. Thus,

if the load vary c ... y and the terminal pressure be suitable for best economy at full the load becomes ... governor would cause the valve gear to cut off steam earlier in t cylinder, which would increase the number of expansions and lower the terminal pressure to such extent as would be unfavorable for economy. Accordingly, it is evident that assigning a proper value for the

terminal pressure at full load is a matter requiring considerable experience and involves considering all the conditions of operation of the engine, present and future. Thus, in addition to the condition already mentioned, if an engine be selected for a rapidly growing plant, a very low terminal pressure could be assigned, that is, an engine too large for the load at the start could be installed, and operated at reduced economy, the loss thus incurred being more than made up when the load has increased by the growth of the plant, thus avoiding excessive overload.

The cut off will depend on the type of engine, a short cut off being employed in stationary practice where ***continuous expansion*** is aimed at, and a late cut off in marine practice where the expansion is interrupted by ***drop***, or ***free expansion*** in the receiver, or receivers.

In fixing the cylinder ratio there are two conditions of operation that would be very desirable, but which conflict; they are:

1. Equal temperature ranges;
2. Equal powers.

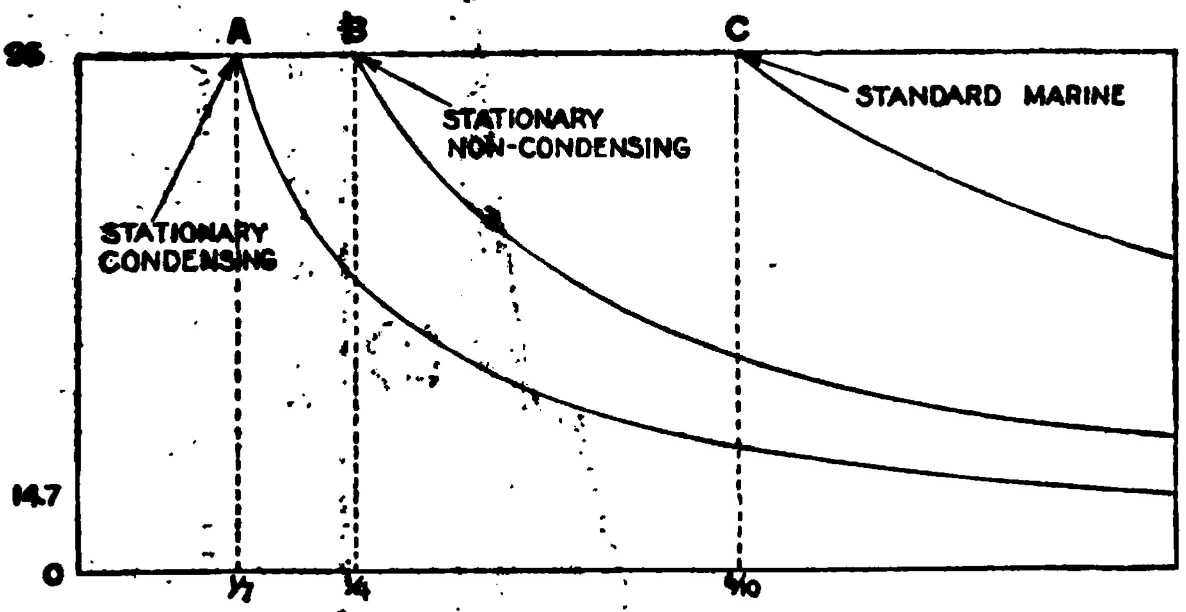


FIG. 1,166.—Typical cut offs. A, 1/7, stationary condensing; B, 1/4, stationary non-condensing; C, 6/10, standard marine.

Thus, the prime object of expanding steam in more than one cylinder being to divide the total range of temperature the ideal condition would obtain when the fall of temperature was equal in each cylinder, as in fig. 1,163. However, in construction, it is desirable that each cylinder develop equal power, as in fig. 1,164.

Comparing the diagrams, it is seen in fig. 1,163, that equal temperature range is obtained by an unequal distribution of power and in fig. 1,164, that equal distribution of power involves unequal temperature ranges.

While the case here given does not show a very marked difference in the temperature ranges for equal work, others would show greater differences especially with triple and quadruple expansion engines.

In designing a compound engine the sizes of the cylinders can be determined with a sufficient degree of accuracy from the theoretical diagram by using a suitable diagram factor. In

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general, the terminal pressure may be safely assumed at from 10 to 12 lbs. for condensing engines and from 20 to 25 lbs. for non-condensing engines.

The following example from Kent will illustrate one method of determining the cylinder diameters for equal horse power.

Example.—Find the cylinder diameters of a compound condensing engine of 2,000 horse power at a speed of 700 feet per minute, with 100 lbs. boiler pressure.

100 lbs. gauge pressure $=115$ lbs. absolute, less drop of 5 lbs. between boiler and cylinder $=110$ lbs. initial pressure absolute. Assuming terminal pressure in *l.p.cyl.* $=6$ lbs., the total number of expansions $=110\div 6=18.33$ Hyp. log $18.33=2.909$. Back pressure in *l.p.cyl.* $=3$ lbs. absolute.

The following formulæ are used in the calculation of each cylinder:

1. Area of cylinder $=\dfrac{\textit{horse power}\times 33{,}000}{\textit{m.e.p.}\times\textit{piston speed}}$

2. ***Mean effective pressure = mean total pressure − back pressure.***

3. ***Mean total pressure = terminal pressure × (1 + hyp. log R).***

4. ***Absolute initial pressure = als. terminal pres. × ratio of expansion.***

Solution.—First calculate the area of ***l.p. piston*** as if all the work were done in the *l.p.* cylinder.

From **3**, mean total pressure $=6\times(1+\text{hyp. log } 18.33)=23.454$ lbs.

From **2**, mean effective pressure $=23.454-3=20.454$ lbs.

From **1**, area of piston $=\dfrac{2{,}000\times 33{,}000}{20.454\times 700}=4{,}610$ sq. ins. $=76.6$ ins. diam.

If half the work or 1,000 horse power be done in the *l.p.cyl.*, the *m.e.p.* will be half that found above, or 10.227 lbs., and the mean total pressure.

$$10.227+3=13.227 \text{ lbs.}$$

From **3**, $1+\textit{hyp. log } R=13.227\div 6=2.2045$

$$\text{or, } hyp.\ log\ R = 2.2045 - 1 = 1.2045$$
$$\text{whence } R \text{ in } l.p.cyl. = 3.335$$

From 4, 3.335×6 = 20.01 lbs. initial pressure in *l.p.cyl.* and terminal pressure in *h.p.cyl.*, assuming no drop between cylinders.

R in *h.p.cyl.* = 110 ÷ 20.01 = 18.33 ÷ 3.335 = 5.497.

From 3, mean total pressure in *h.p.cyl.*

$$= 20.01 \times (1 + \text{hyp. log. } 5.497) = 54.11$$

From 2, 54.11 − 20.01 = 34.1 *m.e.p.* in *h.p.cyl.*

$$\text{From 1, area of } h.p.cyl. = \frac{1{,}000 \times 33{,}000}{700 \times 34.1} = 1{,}382 \text{ sq. in.} = 42 \text{ in. diam.}$$

$$\text{cylinder ratio} = 4{,}610 \div 1{,}382 = 3.34$$

The area of the *h.p.cyl.* may be found more directly by dividing the area of the *l.p.cyl.* by the ratio of expansion in that cylinder, thus

$$4{,}610 \div 3.335 = 1{,}382 \text{ sq. in.}$$

Summary

	High pressure cylinder	Low pressure cylinder
Initial pressure absolute....................	110 lbs.	20.01 lbs.
Terminal pressure absolute..............	20.01 lbs.	16 lbs.
Number of expansions....................	5.497	3.335
Mean effective pressure.....................	34.1 lbs.	18.35 lbs.
Area of piston..............................	76.7 sq. ins.	281 sq. ins.
Diameter of cylinder.....................	11⅞ (approx.)	19 ins. (approx.,
Cylinder ratio..............................	1	3.34

NOTE.—In the above calculations no account is taken of clearance, of compression, of drop between cylinders, nor of area of piston rods. It also assumes that the diagram in each cylinder is the full theoretical diagram, with a horizontal steam line and a hyperbolic expansion line, with no allowance for rounding of the corners. To make allowance for these, the mean effective pressure in each cylinder must be multiplied by a diagram factor, or the ratio of the area of an actual diagram of the class of engine considered with the given initial and terminal pressure, to the area of the theoretical diagram. Such diagram factors will range from .6 to .94.

In a compound engine with a given cylinder ratio, the variable elements of the cycle are, as stated by Heck: 1, load on the engine; 2, total ratio of expansion, as fixed by the *h.p.* cut off; 3, *l.p.* cut off; 4, exhaust pressure.

With the load constant, the work done in the *h.p.* cylinder is increased: 1, by making the *l.p.* cut-off later; 2, by raising the boiler pressure; this slowly decreases *pv.* with amount of steam admitted to perform a given total amount of work per revolution, but it adds to the *h.p.* diagram by raising the admission line and lowering the exhaust line more rapidly than it subtracts by drawing inward the expansion curve. By raising the exhaust pressure: this subtracts more from the bottom of the *l.p.* diagram than is added on account of the increase in *pv.* and the corresponding rise in the receiver pressure.

In a given engine, the preservation of an equal division of load requires that the *l.p.* cut off shall vary with the *h.p.*, in the same direction. If the *l.p.* cut off be left constant, increasing the load increases the share taken by the *l.p.* cylinder.

In proportioning an engine with a view to a certain division of power and a reasonable relative amount of loss by receiver drop, the *h.p.* cylinder must be relatively smaller as the total ratio of expansion is greater. With a fixed *l.p.* cut off and with a constant amount of steam admitted to the *h.p.* cylinder per revolution, the volume of the *h.p.* cylinder can be varied over quite a range with comparatively little effect upon the division of work, but with a great influence upon the receiver drop.

If the *l.p.* cut off be varied in a *proper relation* to the size of the *h.p.* cylinder (that is, so as to keep the proportion of receiver drop about constant), making the latter larger gives it a larger share in the power developed.

Variable Load in Compound Engines.—A compound engine of good design will show considerable economy at full load. If the load vary considerably, the economy is not so good. There are several methods of regulation to meet changes in load, as by

1. Variable cut off;
2. Throttling.

Variable Cut Off.—On most compound Corliss engines the provision for regulation is such that the low pressure cylinder

FIGS. 1,168 and 1,169.—Theoretical combined diagrams of compound engine illustrating *regulation by variable cut off in the h p* pressure cylinder only. AB, full load cut off in *h.p.* cyl. Fig. 1,168 shows effects due to *shortening* the cut off; fig. 1,169, the effects due to *lengthening* the cut off.

can be operated either with a fixed cut off or under control of the governor in any ratio to the cut off in the high pressure cylinder, or a combination of both methods, giving as will be shown, a convenient and accurate control of the receiver pressure, as required for any sort of operating conditions.

As an example of regulating mechanism construction the low pressure valve gear of the Hewes & Phillips cross compound engine is controlled from the governor by the usual cross shaft, in addition to which a differential lever is supplied which is controlled by a quadrant, enabling the relation between the cut off in the *l. p.* and *h. p.* cylinders to be varied. Thus, when the levers are detached by unlocking the bolt which secures them, a different receiver pressure can be carried at will and the load produced by the two cylinders can be equalized, after which the levers can be again attached, locking the bolt by using a knurled thumb nut, when the low pressure valve gear is again under control of the governor. This feature is particularly valuable where engines are lightly loaded when first installed, permitting proper receiver pressure to be carried.

The accompanying diagrams show the effects obtained on the steam and power distribution between the two cylinders by changing the cut off in one or both cylinders.

In figs. 1,168 and 1,169, let ABCD, and DCEFG, be diagrams of a compound engine running at full load with continuous expansion, that is without fall of pressure in the receiver, and suppose the regulation to be variable cut off in the *h. p.* cylinder only.

Now, if the load decrease, the governor will shorten the cut off from B to some point *b*, changing the expansion line from BCE, to *b c′ e*, and lowering the terminal pressure from FE to F *e*.

The cut off being fixed in the low pressure cylinder it will be on the new expansion line at *c*, vertically below C.

Shortening the cut off has evidently caused the *h. p.* diagram to loose the area *b* B C *c′*, and gain the area D *c′ cd*, giving the area M, or A *b c d*; the low pressure diagram has lost both the areas C E *e c*, and D C *c d*, giving the area S, or *d c e* F G. It will be noted that *h. p.* diagram has gained in height a distance D *d*, while the *l. p.* diagram has lost in height by the same amount; clearly the area of the *h. p.* diagram has increased with respect to the area of the *l. p.* diagram.

It follows from the explanation just given that shortening the cut off in the high pressure cylinder only

1. *Brings a larger proportion of the load on the high pressure cylinder;*

2. *Increases the number of expansions in the high pressure cylinder;*

3. *Reduces the receiver and terminal pressures;*

4. *Increases the total number of expansions.*

Fig. 1,170.—Actual combined diagram illustrating regulation by variable cut off in the *h.p.* cylinder only. As the cut off is lengthened the total power of the engine is increased and the larger share of the increased power being carried by the *l.p.* cylinder. With an early cut off and at low powers, the larger share of the work is done in the *h.p.* cylinder, and as this power is reduced to a minimum the power in the *l.p.* cylinder may be reduced to zero.

Fig. 1,171.—Actual combined diagrams illustrating effect of varying the cut off in the *l.p.* cylinder. In the figure it is seen that shortening the cut off in the *l.p.* cylinder throws a larger share of the total work into the *l.p.* cylinder. Thus when *l.p.* cut off is changed from one-half to one-quarter, the areas of the diagrams will be changed, the *h.p.* diagram being reduced in area and the *l.p.* diagram increased, as shown.

The effects of lengthening the cut off in the *h.p.* cylinder only are shown in fig. 1,169. Here the diagrams ABCD and DCEFG, correspond to full load as in fig. 1,168. Now if the

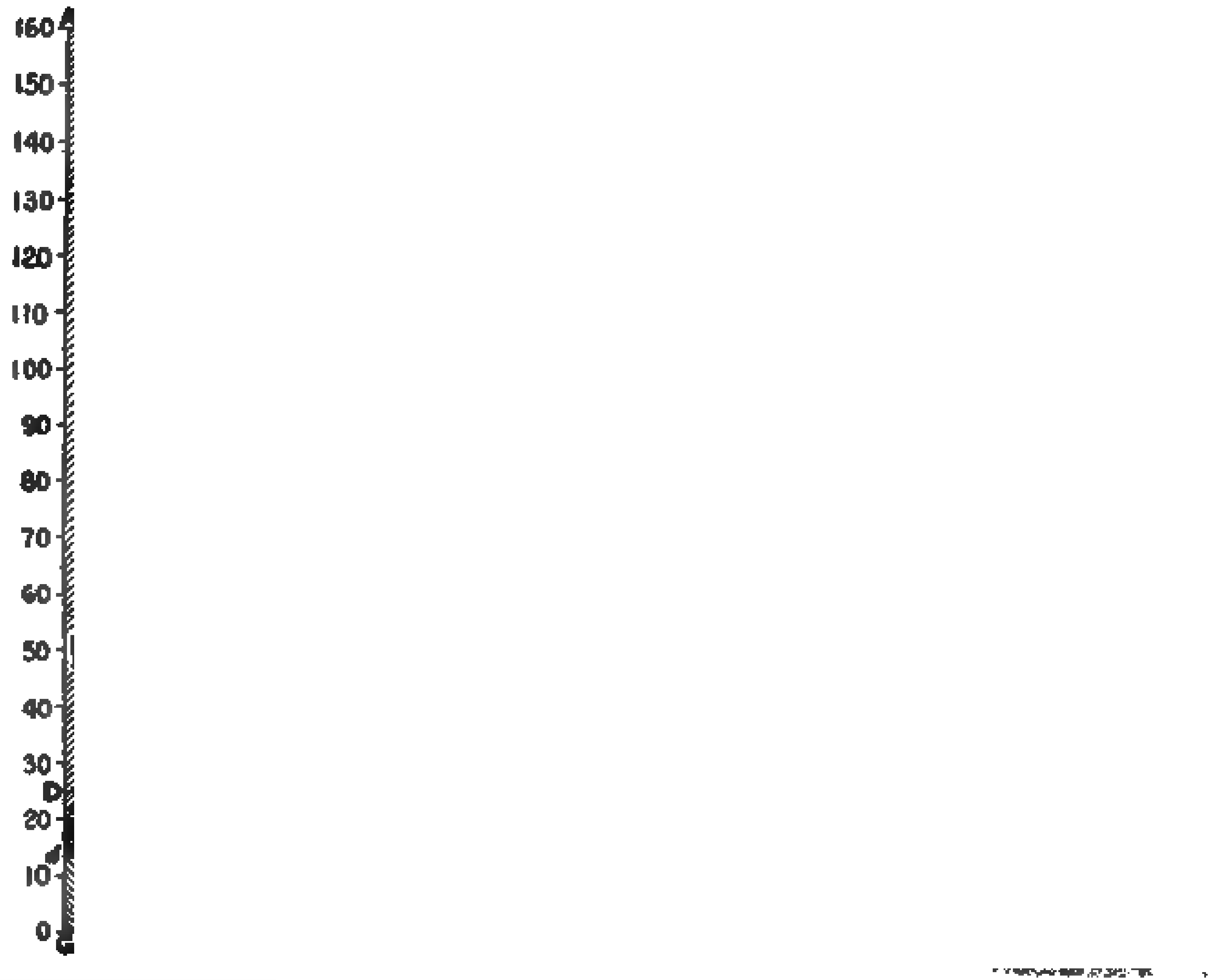
load increase the governor will lengthen the cut off from B, to some point *b*, giving the new diagram **M**, or *Abcd* and **S**, or *dce*FG. From the figure it is clear that lengthening the cut off in the *h.p.* cylinder only:

1. *Brings a larger proportion of the load on the low pressure cylinder;*

FIG. 11,772.—Regulation by variable cut off in *h.p.* cylinder and constant cut off in *l.p.* cylinder. With equal work division for the middle curves L,F, it is evident that "the *h.p.* cylinder takes a *larger* share of a smaller load (shortened *h.p.* cut off), as per curves M,S, and a *smaller* share of a larger load (lengthened *h.p.* cut off), as per curves M',S'."—*Heck.*

2. *Reduces the number of expansions in the high pressure cylinder;*
3. *Increases the receiver and terminal pressures;*
4. *Reduces the total number of expansions.*

A second method of regulation and the one generally used is to vary the cut offs in both cylinders, the *l.p.* cut off varying less



FIGS. 1,173 and 1,174.—Theoretical combined diagrams of compound engine illustrating *regulation by variable cut off in both cylinders*. In the figures AB, and DC, are full load cut off. Fig. 1,173, shows effects of *shortening* the cut off; fig. 1,174, effects of *lengthening* the cut off.

rapidly than the *h. p.* cut off. The effect of this regulation being shown in figs. 1,173 and 1,174.

For comparison the same full load diagrams ABCD and DCEFG, are used as in figs. 1,168 and 1,169. In fig. 1,173 suppose first the *h. p.* cut off be shortened from B, to *b*.

If the *l. p.* cut off had remained fixed, the same diagram A *b c d* and *d c e* F G, would be obtained as in fig. 1,169, increasing the proportion of load carried by the *h. p.* cyl.

Now suppose the *l. p.* cut off be shortened to some point *c''* (which will fall on the expansion line through *b*). The effect of this is to increase the area of the *l. p.* diagram and reduce the area of the *h. p.* diagram,thus tending to equalize the distribution of work. Because of the absence of drop, it causes the expansion to proceed in the *h. p.* cylinder below receiver pressure from *c''*, to *c*, resulting in negative work represented by the solid black triangular area *c'' c' c*, which must be subtracted from A *b c'' d'*, to obtain the effective or net area of M. The effect of this would be to cause steam to blow back from the receiver to the *h. p.* cyl. at release, (tending in the case of a Corliss engine to lift the exhaust valve from its seat) till the pressure was equalized or raised from *c*, to *c'*, and causing the *h. p.* piston to oppose or "buck" the *l. p.* piston from *c''*, to *c'*, the end of the *h. p.* stroke.

Shortening the cut off in both cylinders then (when there is no drop)

1. *Tends to equalize the loads carried by the two cylinders;*
2. *Tends to equalize the expansions in the two cylinders;*
3. *Tends to maintain a constant receiver pressure;*
4. *Reduces the terminal pressure;*
5. *Increases the total number of expansions.*

Now, with the same method of regulation, suppose both cut offs be lengthened as in fig. 1,174, the *h. p.* cut off from B, to *b*, and the *l. p.* cut off from *c*, to *c''*.

The corresponding diagrams are M, or A *b c c' d*, and S, or *d c'' e* FG. The expansion is interrupted at *c*, and the pressure suddenly falls or "***drops***" to *c'*, due to lengthening the *l. p.* cut off. This lengthening of the *l. p.* cut off causes the *l. p.* diagram to lose the cross shaded area *d' c c'' d*, and the *h. p.* card to gain the area *d' c c' d*, while the solid black triangular area *c c' c''*, is lost to both diagrams, thus tending to equalize M and S, or the work done in the two cylinders.

FIGS. 1,175 and 1,176.—Theoretical combined diagrams of compound engine illustrating regulation by throttling and combined throttling and variable cut off; fig. 1,175, *condensing operation;* fig 1,176, *non-condensing operation.*

Throttling.—Where a load, considerably less than the full load, is to be carried for an extended period, as when a large engine is installed in a growing plant, and must operate for some time with a load so small as to be beyond the economical range of the engine, the economy may be improved by reducing the boiler pressure or by throttling either by hand or governor control.

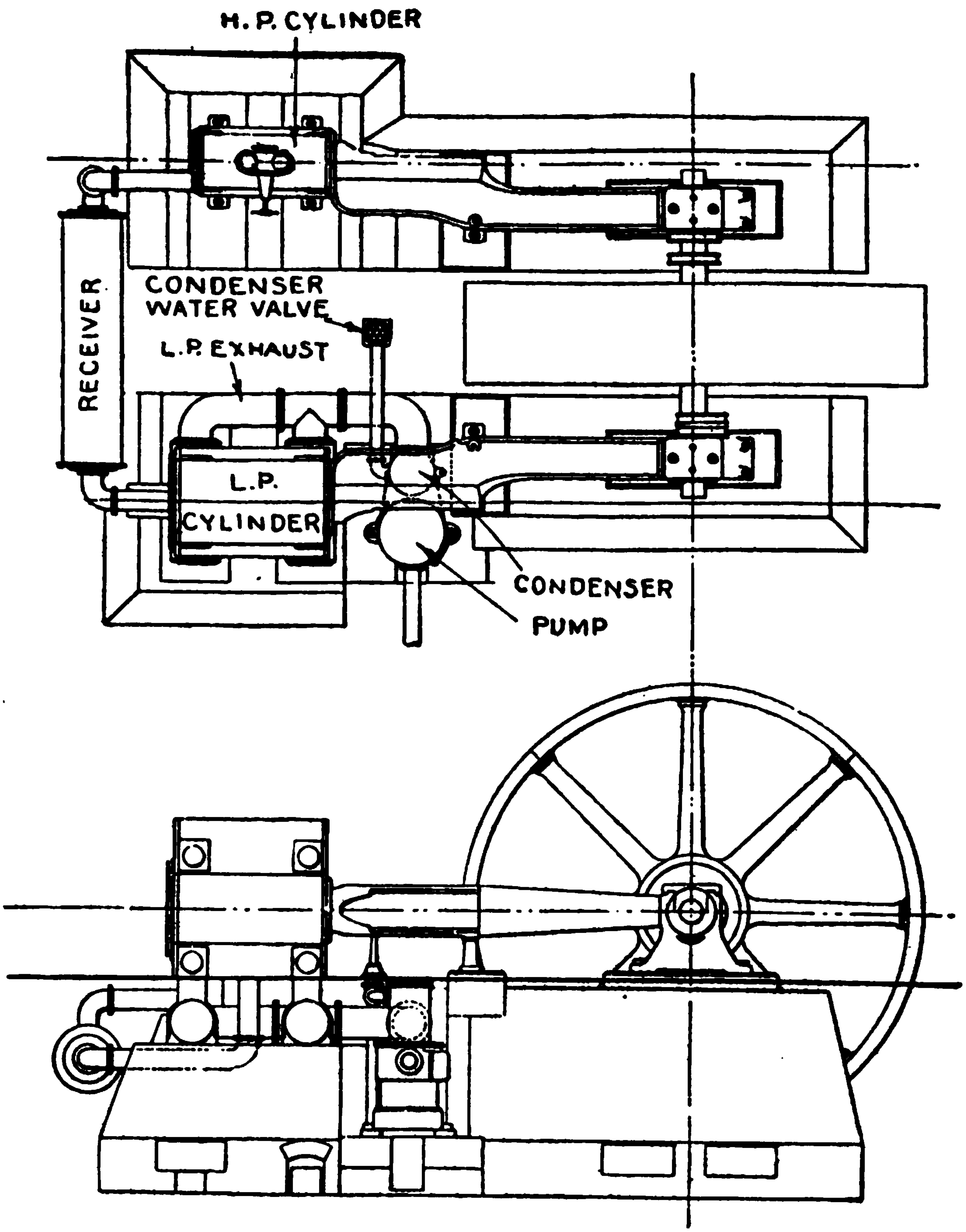
To illustrate, suppose, as before in fig. 1,175 that ABCD and DCEFG, are the full load diagrams for which the engine was designed.

FIG. 1,177.—Actual combined diagrams illustrating regulation by throttling.

Since the load for the first few months or year is to be very light, it might be necessary to reduce the initial pressure to, say, 80 lbs. if throttling alone were used for regulation, giving the diagrams *abcd* and *dce*FG. Such a method might give a terminal pressure too low for best light load economy.

A combination of throttling and variable cut off would give better results, that is, the pressure might be throttled down to Ga', and the cut off lengthened to b' (corresponding to equal diagram area), thus raising the terminal pressure for a given load. If the *l.p.* cut off were not changed, the new diagrams would be M and S, or $a'b'c'd'$ and $d'c'e'$FG, respectively.

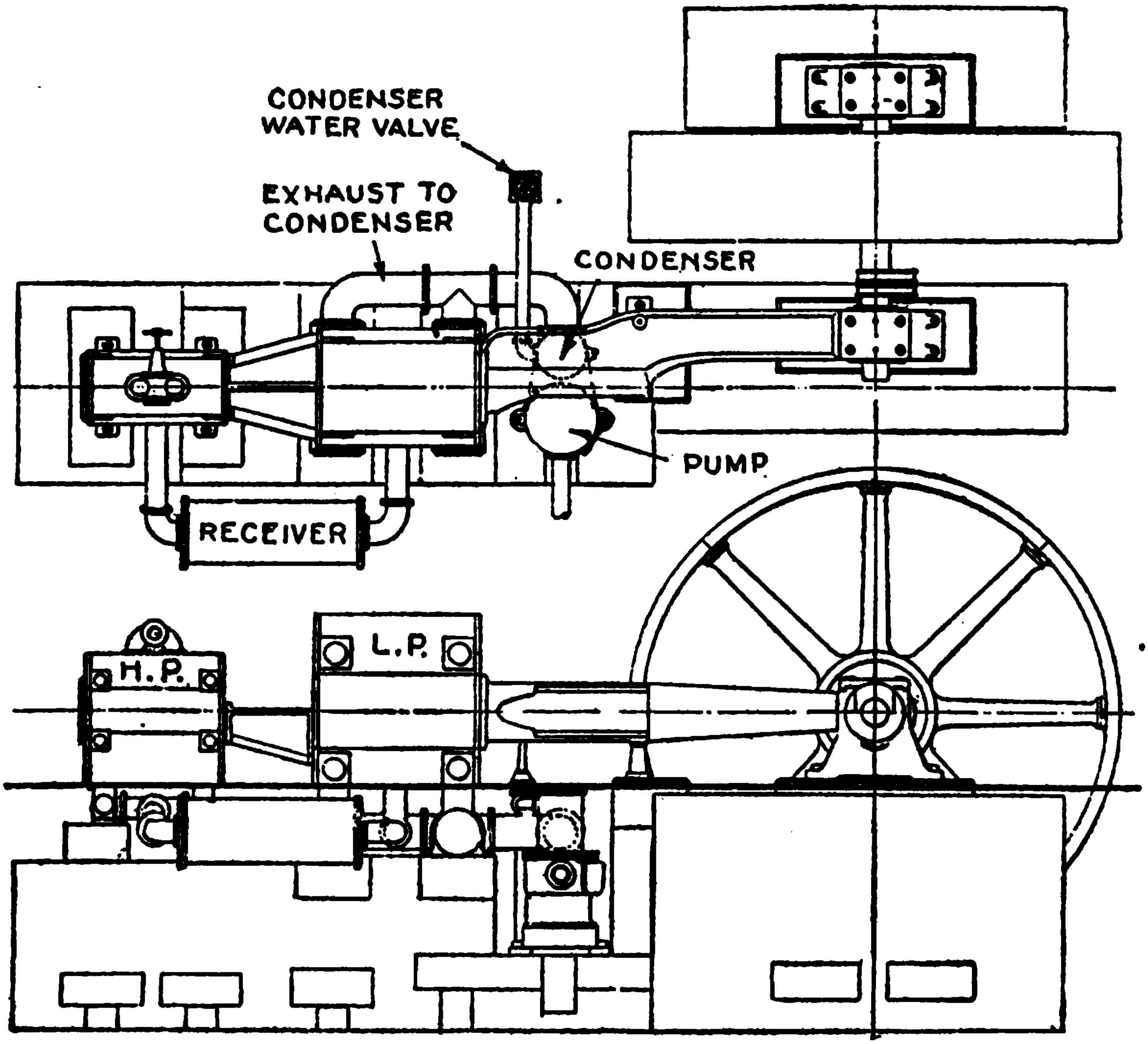
In the case of a non-condensing engine as in fig. 1,176, let ABCD and DCEFG, represent the full load diagrams. If throttling alone be employed for regulation, when the load is very light it may be necessary to so reduce the initial pressure that the expansion would proceed until the pressure in the cylinder became less than the exhaust pressure, giving diagram such as *abcd* and *dce*FG, with the resulting loop of negative



FIGS. 1,178 and 1,179.—Plan and elevation of ***cross compound*** engine. Engines of this type are arranged similar to a form of simple engines, each cylinder working on its own crank, set at right angles to the other, and each having its own frame, cross head, connecting rod, etc. This construction enables one side to be run as a simple engine, in case of accident to the other, and the two cranks give a nearer uniform turning motion as compared with the tandem arrangement. A feature of the cross compound type is that one side may be installed at a time, in case the full power of the engine be not needed at once, the *h. p.* side being put in first, and the wheel being made large enough to drive the full power of both cylinders.

work hef'', shown in solid black section. This may be avoided by still further reducing the initial pressure and lengthening the *h. p.* cut off obtaining diagrams such as M and S, or $a' b' c' d'$ and $d' c' ff' g$.

The diagrams show that non-condensing compounds are most suitable when the load is fairly constant, and because of the liability of expanding below exhaust pressure with the accompanying loop of negative work, they should not be operated with a terminal pressure lower than about 20 lbs., as this allows a little margin for variation in load.



FIGS. 1,180 and 1,181.—Plan and elevation of tandem compound engine. ***In arrangement***, both cylinders have the same axis, being attached to one piston rod and working upon one crank. The *h. p.* cylinder is usually placed behind the *l. p.*, the latter being attached to the frame, and sufficient space being left between the cylinder to allow the *l. p.* piston to be withdrawn, the piston rod being arranged to uncouple for that purpose. This arrangement is best where-ever the width available is limited, or wherever the fly wheel or driving pulley must be placed alongside of wall Sometimes for large powers, two tandem engines are arranged as a pair, to secure ability to run either side alone and giving smaller cylinders than would be necessary in a cross compound engine of the same power.

Combined Diagrams of a Compound Engine.—According to Prof. Kennedy, "the only way of correctly combining the diagrams of a compound or other multi-stage expansion engine is to set off all the diagrams to the same horizontal scale of volumes, adding the clearances to the cylinder capacities proper. Where this is attended to, the successive diagrams fall exactly into their right places relatively to one another, and would compare properly with any theoretical expansion curve." This method of combining diagrams is the one commonly adopted but has been criticised as being inaccurate.

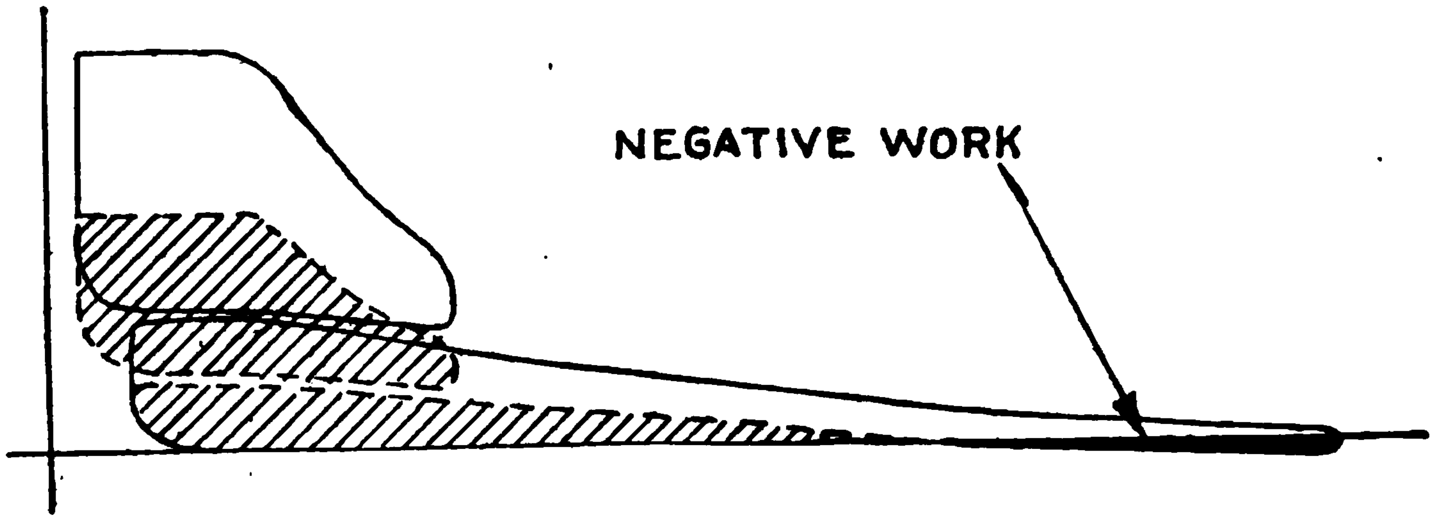


FIG. 1,182.—Actual combined diagrams illustrating regulation by throttling in *non-condensing* compound engine. The solid black section shows a loop of negative work caused by too light load, the steam expanding below the pressure of the exhaust.

The method is shown in fig. 1,183. Take a length of say, 12 ins. for the length of the *l.p.* diagram, set off on a horizontal line which will serve both as a scale of volume and as the absolute zero line pressure. Reproduce the *l.p.* diagram from the original by drawing ordinates and laying off points to the new scale. Now lay off the clearance of the *l.p.* cylinder AS.

At A, draw the perpendicular OA, or clearance line giving O, the point of zero volume. The common scale of pressure may be that of the original *l.p.* diagram or any convenient multiple.

To reproduce the *h.p.* diagram to the new scales, set off the clearance volume and piston displacement of the *h.p.* cylinder to the same scale of volume as used for the *l.p.* cylinder, and divide the piston displacement into ten equal parts. Then transfer to these division lines, the absolute

forward and back pressure given at the corresponding division lines of the original high pressure indicator diagram.

The original indicator diagrams should first be measured with the scale of the spring used in taking the diagram, and the actual absolute forward and back pressures marked upon them, so that these numbers can be transferred at once to the combined diagram with the enlarged scale of pressures.

The saturation curve is transferred from the original diagram to the combined diagram in each case, by the method of "dry steam fraction."

It is probable that the saturation curve of the respective cylinders will not coincide—that is, will not be continuous. This could only occur if the same weight of cushion steam was retained in each cylinder. Generally

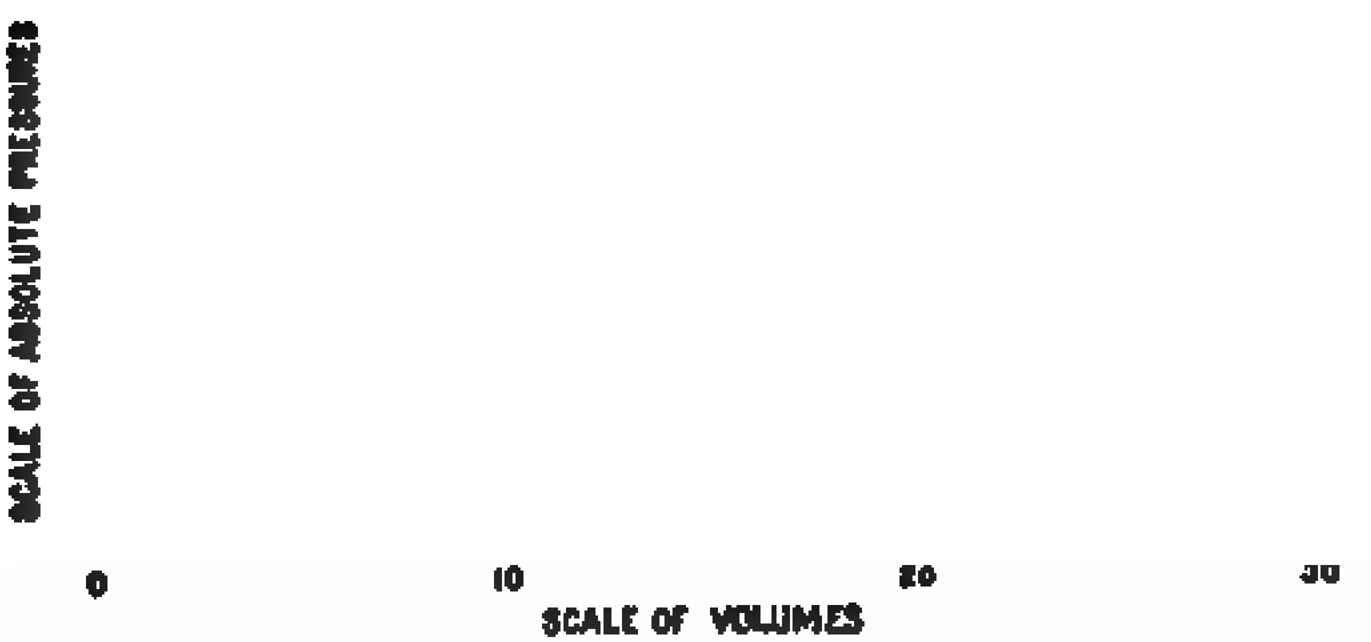


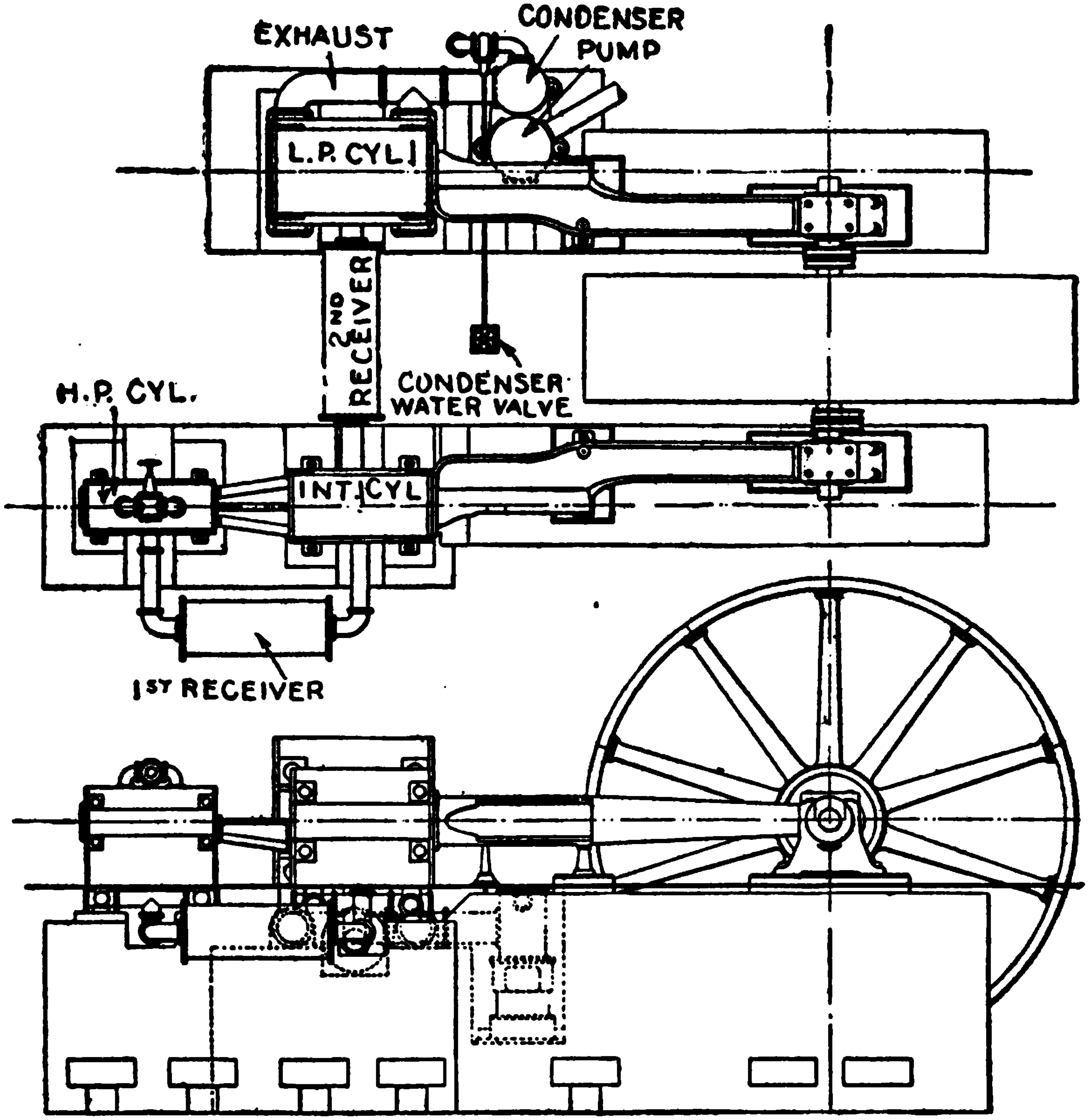
Fig. 1,183.—Combined diagrams of a compound engine. ***In combining the diagrams,*** they are first reduced in length to relative scales that correspond with the relative piston displacements of the cylinders. The diagrams are then placed at such distances from the clearance line of the proposed combined diagram as to represent correctly the clearance in each cylinder.

the weight of cushion steam is less in the lower pressure cylinders, and therefore the saturation curve of the first cylinder falls outside that of the second cylinder.

Triple Expansion Engines.—When the conditions of operation are such that the load is constant, or varies but slightly, and there is condensing water, very high economy can be obtained from the triple expansion engine. In order to obtain the maximum economy all the conditions under which an engine

of this type is to work must be fully considered, and then only can the proper arrangements of cylinders be decided upon.

The two arrangements in extensive use for the cylinders of triple expansion engines are:

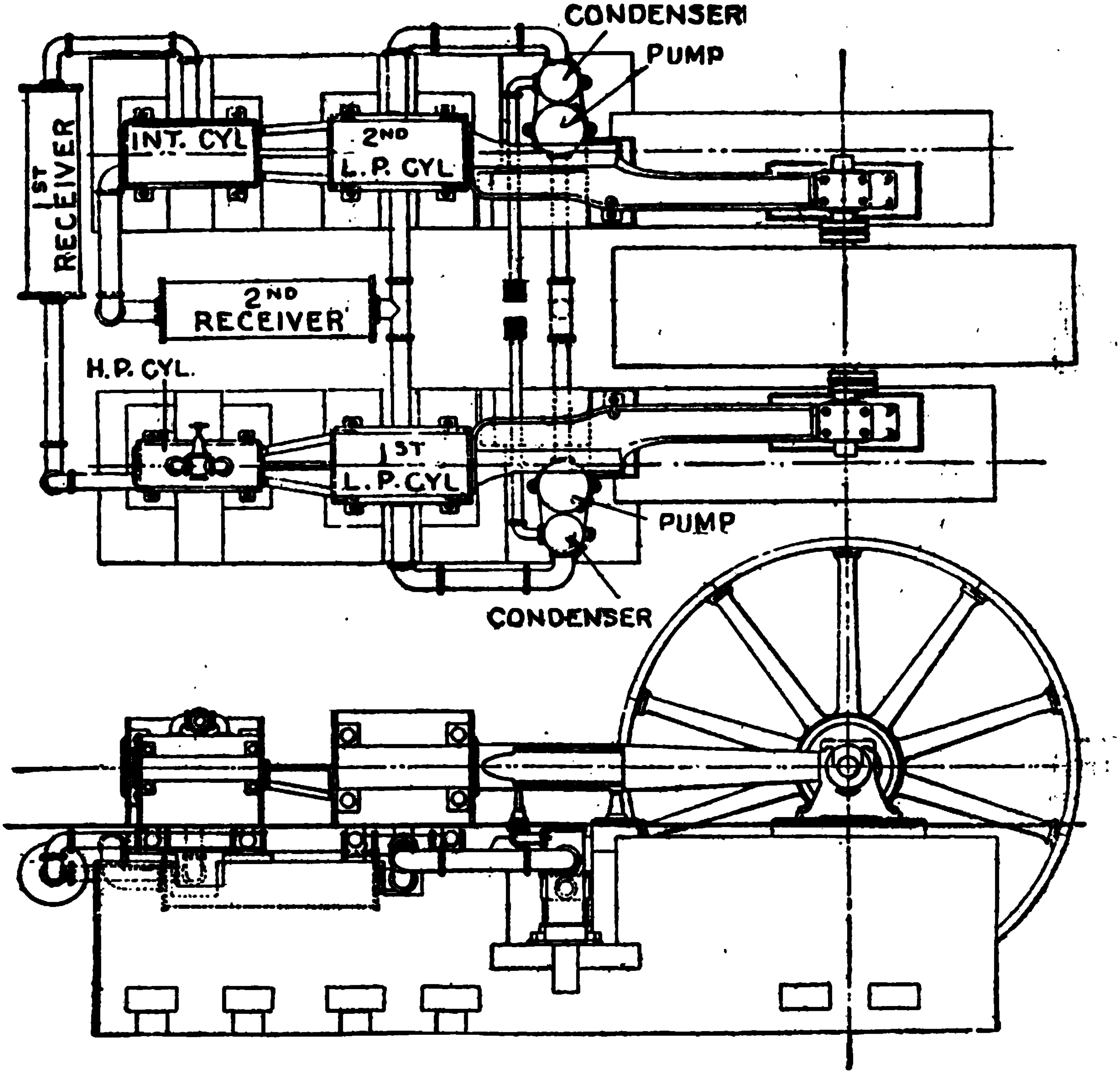


FIGS. 1,184 and 1,185.—Plan and elevation of three cylinder triple expansion condensing engine arranged with *h. p.* and *int.* cylinder tandem and *l. p.* cylinder with separate crank at 90°.

1. The high pressure and intermediate cylinders are arranged tandem, working on one crank, the low pressure cylinder working

on another crank placed at right angles with the first, as shown in figs. 1,184 and 1,185. With this arrangement one side may still be run in case of accident to the other.

2. For large powers the low pressure cylinder may be divided

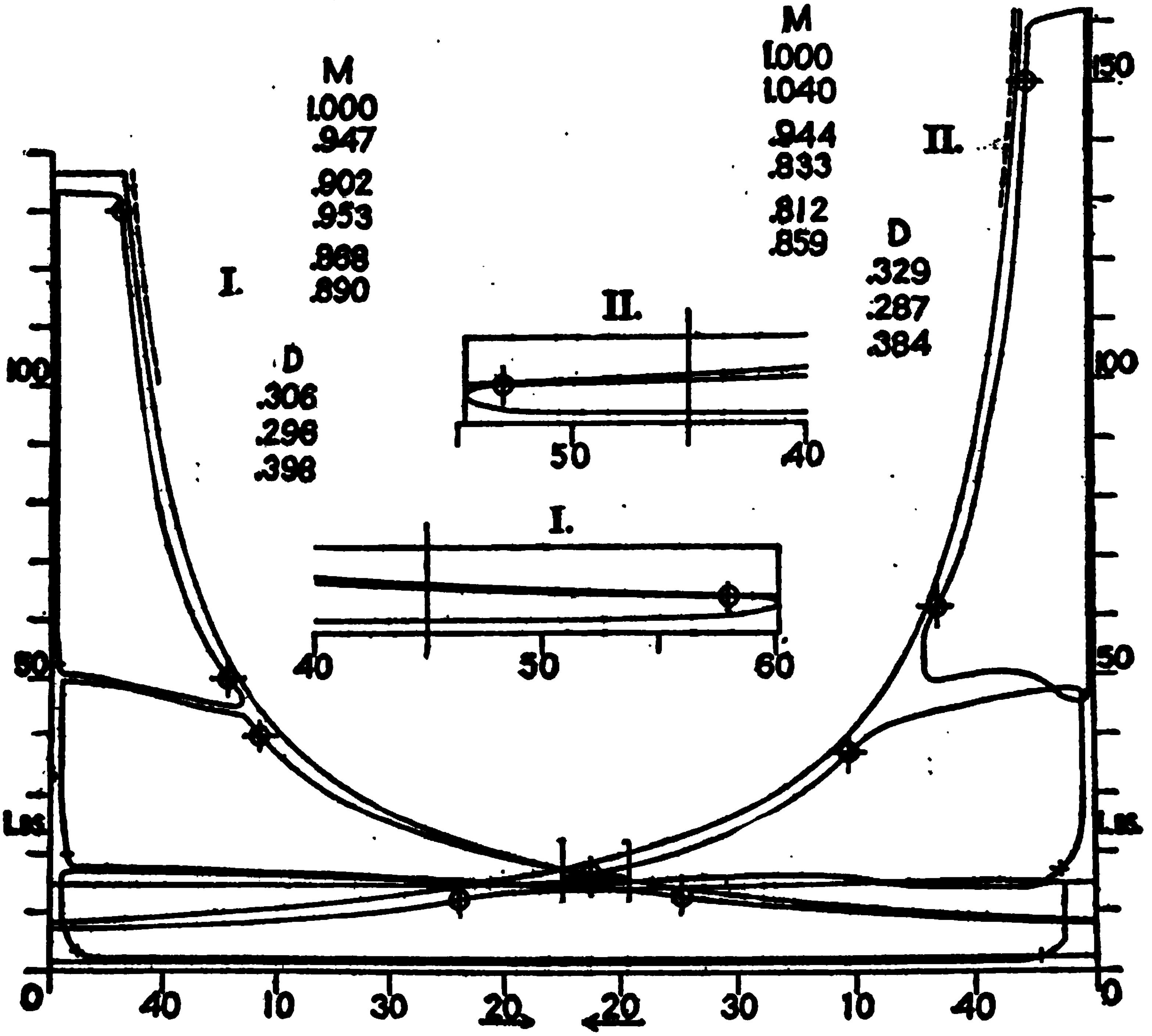


FIGS. 1,186 and 1,187.—Plan and elevation of four cylinder, triple expansion, condensing engine, arranged in double tandem: *h. p.* and first *l. p.* cylinders in one pair and *int.* and second *l. p.* cylinders in another, the two pairs operating cranks at 90° as shown.

into two in order to reduce its size, the arrangement thus being similar to a pair of tandem engines.

Each *l.p.* cylinder is bolted to a frame and works on its own crank, and has a *h.p.* or *int.* cylinder, as the case may be, arranged behind it on the same center line.

The usual method of regulation consists of variable cut off in the *h.p.* cylinder under control of the governor, the cut off of the other cylinders being adjustable by hand according to the cylinder ratios. In order to



FIGS. 1,188 and 1,189.—Two combined cards of vertical triple expansion engines with reheaters and jackets. Fig. 1,188, Allis-Chalmers engine at Milwaukee. Cylinders 28, 48 and 74 by 6 ; *r.p.m.* 20.3; indicated horse power 573.9; steam per horse power hour 11.8. Fig. 1,189, Snow engine at Indianapolis; cylinders 29, 52 and 80 by 60; *r.p.m.* 21.2; indicated horse power 782.9; steam per horse power hour 11.5.

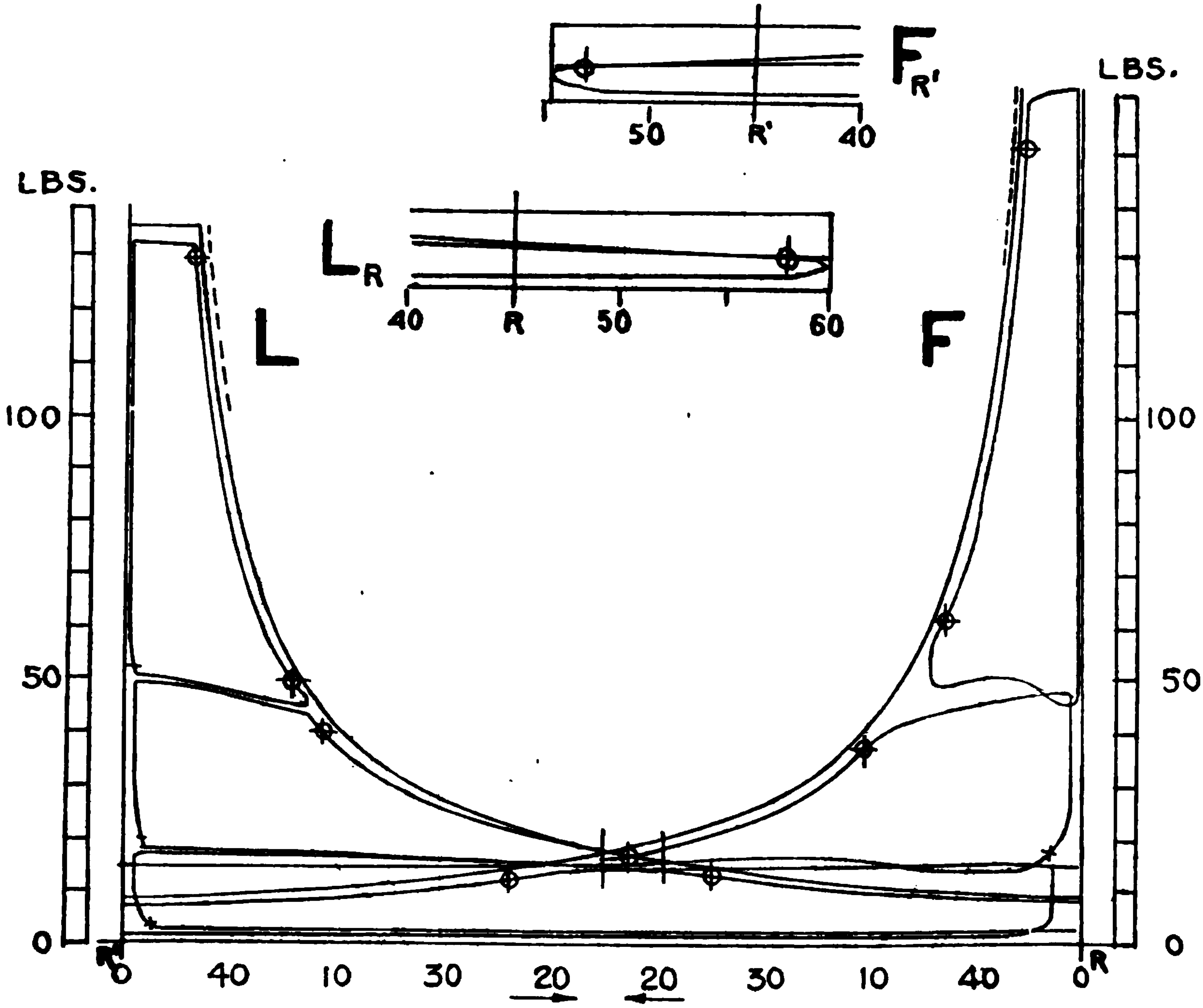
avoid a drop in the expansion, a means is usually provided for bringing the cut off of the other cylinders under governor control when necessary.

The following example from Tribe will illustrate one method of figuring the cylinder diameters of a triple expansion engine.

lines AB, CD, EF, at such distances from OY, as to represent respectively the clearance volume in the *h.p.*, *i.p.*, and *l.p.* cylinders, taking care to select the scale of volume such that the *l.p.* volume plus its clearance as measured between the lines OY and GK, will come within the desired limits of the diagram. 4, Lay off on each card the line of zero pressure or the perfect vacuum line, as shown by OX, in the small diagram A. 5, Take next the *h.p.* card as at A, for example, and select any point such as P. Measure in any convenient units the distances MP and MN; multiply the volume of the cylinder by the former and divide by the latter. This will give the volume swept in the *h.p.* cylinder from the beginning of the stroke to the point P. 6, The corresponding point P, of the combined diagram is then found by measuring from AB, a distance HP, representing this volume, and from OX, a distance JP, representing the pressure PQ, on the card. This will give the point P, and other points are found in a similar manner, as many as may be needed to determine the form of the card as shown. It is to be especially noted that the *h.p.* card of the combined set is the same as that at A, but drawn simply with different scales, and therefore more or less distorted in appearance. 7, The points necessary to determine the other cards of the combination are found in a precisely similar manner, remembering that in each case volume is measured from the clearance line CD, or EF, while the pressure must be measured from the line of zero pressure for the card and laid off from the corresponding line OX, of the combined set. This diagram shows the general manner in which the steam expands on its way through the engine. An expansion line PQ, shows the general law of expansion as a continuous operation. PR, is an ideal expansion line laid down as a hyperbola, all points in the curve corresponding to the condition that the product of volume by pressure shall be constant, or in symbols, $pv = constant$. This shows the result of the so called true hyperbolic expansion law, and as appears from the diagram, the actual expansion line is somewhat below this ideal line. The equation to the actual expansion line may be expressed in the form $pv^n = constant$, where n is an exponent having values usually lying between 1.15 and 1.2. The equation $pv^{1.15}$ may be taken as very commonly representing this line in good average practice. The extent to which the area bounded by the line PR, and the clearance lines on the left is well filled in, is an indication of the degree to which the performance of the actual engine approaches that of an engine having true hyperbolic expansion. The relation between the actual engine and such an ideal case is usually expressed by a percentage factor known as the "card factor." For good practice with triple expansion engines, this factor will be found from .6 to .7. With quadruple expansion engines representative values are found from .55 to .6.

Figs. 1,191 and 1,192.—Diagrams from compound locomotives. Fig. 1,191, cross compound (two cylinder) consolidation type; cylinders 23 and 35 by 32. Fig. 1,192, De Glehn compound (four cylinder) Atlantic type. In fig. 1,192 the *l.p.* cut offs are considerably less than the *h.p.*, which lowers the receiver pressure. Here, according to Prof. Heck, the advantage of the work division is not apparent, unless it be desired to diminish the amount of work done by the *l.p.* cylinders, which are between the frames and act upon crank pins of limited area. The peculiar shape of the *h.p.* exhaust line in fig. 1,192, for an engine with cranks opposite, is due to the fact that the receiver is common to the two sides of the locomotive, the use toward mid-stroke being carried by inflow from the other *h.p.* cylinder.

NOTE.—***Nordberg Pumping Engine at Wildwood, Pa. Eng. News.*** May 4, 1899, Aug. 23, 1900. *Trans. A. S. M. E.* 1899. The peculiar feature of this engine is the method used in heating the feed water. The engine is quadruple expansion, with four cylinders and three receivers. There are five feed water heaters in series *a*, *b*, *c*, *d*, *e*. The water is taken from the hot well and passes in succession through *a* which is heated by the exhaust steam on its passage to the condenser; *b* receives its heat from the fourth cylinder and *c*, *d* and *e* respectively from the third, second and first receivers. An approach is made to the requirement of the Carnot thermodynamic cycle, *i. e.*, that heat entering the system should be entered at the highest temperature; in this case the water receives the heat from the receivers at gradually increasing temperatures. The temperatures of the water leaving the several heaters were, on the test, 105°, 136°, 193°, 260° and 311° F. The economy obtained with this engine was the highest on record at the date (1900) viz. 162,948,824 ft. lbs. per million *B.t.u.*, and it has not yet been exceeded (1909).—*Kent.*



FIGS. 1,193 and 1,194.—Combined card for two vertical triple expansion pumping engines with reheater and jackets. Diagrams L, are from Allis-Chamlers engine at Milwaukee; diagrams F, from Snow engine at Indianapolis. L_R, is a continuation of *l.p.* diagram L, and $F_{R'}$, a continuation of *l.p.* diagram F.

NOTE.—***Triple expansion pumping engines.*** Prof. Denton says: "Pumping engines in the United States have been developed in the triple expansion fly wheel type to a degree of economy superior to that afforded by any compound mill or electric engine, and, for saturated steam, superior to that of the pumping engines of any other country. This is because their slow speed permits of greater benefit from jackets and reheaters and of less losses from wire drawing and back pressure. These causes, together with the greater sub-division of the range of expansion, have resulted in records made between 1894 and 1900 of 11.22, 11.26 and 11.05 lbs. of saturated steam per *i.h.p.*, with 175 lbs. steam pressure and from 25 to 33 expansions, in the cases of the Leavitt, Snow and Allis pumping engines, respectively the corresponding heat consumption being by different dispositions of the jacket drainage, 204, 208 and 212 thermal units per *i.h.p.* minute; while later the Allis pump, with 185 lbs. steam pressure, has lowered the record to 10.33 lbs. of saturated steam per *i.h.p.*, with 196 *b.t.u.* per *h.p. minute.*"

NOTE.—***At the present time*** the figures given above for triple expansion pumping engines do not represent any remarkable degree of economy when compared with locomobile and uniflow engine performance. It should be noted that the load condition is very favorable for high economy in pumping engines, as in most cases the load is constant, hence with properly proportioned cylinders full load operation is obtained.

FIG. 1,195.—Combined cards from Nordberg quadruple expansion pumping engine, from test by Prof. R. C. Carpenter. In this Nordberg pumping engine, steam is taken from the expansion side of the cycle at a number of points, and used to heat the feed water by successive steps. The water first passes through a surface heater in the exhaust pipe, between the *l.p.* cylinder and the condenser; then it goes through a series of mixing heaters, being pumped from each lower one to the next one, in which there is a higher temperature and pressure. Steam for these heaters is taken from the exhaust pipe, from the *l.p.* cylinder at release, from the third, the second, and the first receivers—the jacket and reheater drains being included in the steam and water thus abstracted; at the *l.p.* release the steam for the heater escapes from the cylinder through a small special valve which is opened for a little while near the end of the stroke. Then the boiler, instead of being obliged to heat the feed water from the exhaust temperature, receives this water at a temperature near to that in the first receiver.

Example.—Find cylinder diameter of a 774 horse power triple expansion pumping engine, with reheater in the second receiver only, to operate at 155 lbs. gauge pressure; 6.23 lbs. abs. terminal pressure, 4 lbs. abs. back pressure, piston speed 215 ft. Assumed losses: drop between boiler and engine 2 lbs.; in 1st receiver 3 lbs.; in 2d receiver ½ lb.

$$155+15-2=168 \text{ lbs. abs.} \quad 168\div 6.23=27 \text{ total expansions}$$

$$\sqrt[3]{27}=3 \text{ expansion in each cylinder}$$

$$1+\text{hyp. log of } 3=2{,}0986.$$

High pressure cylinder.

Initial pressure 168 lbs., $\frac{168\times 2.0986}{3}=117.5$ forward mean pressure.

$168\div 3=56$ back pressure, and $117.5-56=$ ***61.5 M.E.P.***

$$\frac{33{,}000\times 258}{61.5\times 215}=643 \text{ sq. ins. area.}$$

add area of piston rod, and ***high pressure cyl. = 29″ diameter.***

Intermediate Cylinder.

Assumed drop without reheater 3 lbs.; initial pressure $=56-3=53$ lbs.

$$\frac{55\times 2.0986}{3}=37 \text{ formed mean pressure.}$$

$53\div 3=17.7$ back pressure, and $37-17.7=$ ***19.3 M.E.P.***

$$\frac{33{,}000\times 258}{19.3\times 215}=2{,}052 \text{ sq. in. area.}$$

add area of piston rod, and ***intermediate cyl. = 52″ diameter.***

Low pressure cylinder.

Assumed drop with reheater ½ lbs.; initial pressure $17.7-.5=17.2$ lbs.

$$\frac{17.2\times 2.0986}{3}=12 \text{ lbs. forward mean pressure.}$$

Assumed back pressure 4 lbs., and $12-4=$ ***8 lbs. M.E.P.***

$$\frac{33{,}000\times 258}{8\times 215}=4{,}950 \text{ sq. in. area.}$$

add area of piston rod and ***low pressure cyl. = 80″ diameter.***

Quadruple Expansion Engines.—The two important conditions for operation of quadruple expansion engines are:

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CHAPTER 20

PUMPING ENGINES

The term pumping engines relates to large pumps, as those used for city water works, in distinction from small pumps such as boiler feed pumps, etc.

Early History.—The history of the pumping engine is virtually the history of the steam engine, for originally, and for many years the only way in which the steam engine was utilized was for pumping water out of the coal mines of England.

In 1698 Capt. Thomas Sevary secured Letters Patent for a machine for raising water by steam. This engine was used extensively for draining mines and the water was, in some instances, made to turn a water wheel, by which lathes and other machinery were driven.*

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*NOTE.—Capt. Sevary's pump consisted of two boilers, and two receivers for the steam together with valves and the needful pipes. One of the receivers being filled with steam, its communication with the boiler was then cut off, and the steam condensed with cold water outside of it. Into the vacuum thus formed the atmosphere forced the water from below, when the steam was again caused to press upon the water and drive it still higher.

NOTE.—In 1641 a Florentine pump maker constructed an atmospheric or "sucking" pump, and tried to raise water to a height of 50 or 60 feet. The attempt proving a failure, though an examination showed the pump to be perfect, the difficulty was submitted to Galileo, a native of Florence. The action of such pumps was in those days explained on the principle that nature "abhorred a vacuum," and by some occult means tried to prevent such being formed, or to fill it, if formed, with whatever was most convenient. The limit of this occult power was not surmised, and Galileo, then an old man of 80, did nothing more to solve the difficulty than to state that this law was "limited, and ceased to operate for heights above 32 or 33 feet." Torricelli, in 1643, announced his great discovery that water is raised in pumps by the pressure of the air, and by refined experiments he determined the intensity of that pressure. Pascal, shortly afterwards, silenced objectors to the new law by showing that the height of the barometric column was, in accordance with Torricelli's discovery, less at the top of a mountain than at its base.

In 1705, Thomas Newcomen, with his associates, patented an engine which combined, for the first time, the cylinder, piston, and separate boiler. This soon became extensively used for draining mines and collieries, and the engines grew to be of gigantic size, with cylinders 60 inches or more in diameter, and other parts in proportion.

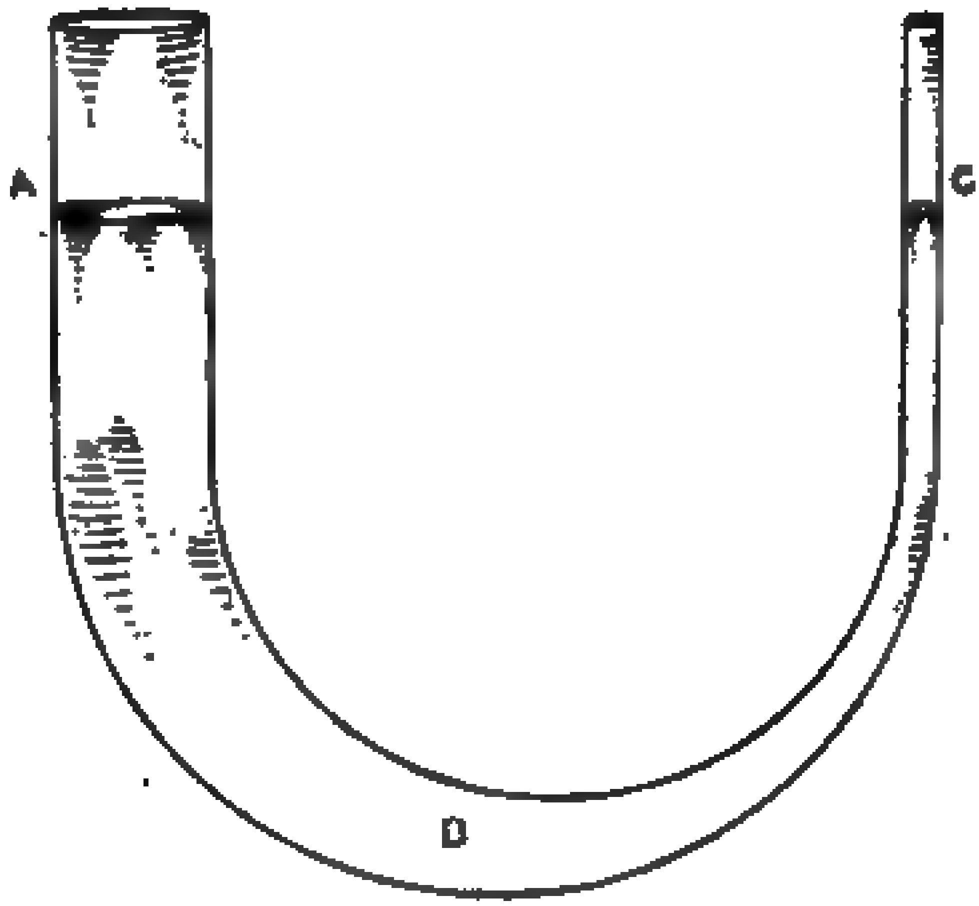
This engine was, in course of years, used in connection with the Cornish Pump, whose performance in raising water from mines came to be a matter of the nicest scientific investigation, and adopted as the standard for the duty or work, by which to compare the multitudinous experimental machines introduced from the time of Watt and to that of Corliss.

FIG. 1,197.—The first pumping engine used in New York City. It worked on the principle of the Newcomen atmospheric engine.

Hydraulics.—Preliminary to the consideration of pumping engines it is essential that the reader be familiar with a few principles of hydraulics as here presented:

The term *hydraulics* is commonly, though ill advisedly, defined as *the science which treats of liquids, especially water in motion*. **Properly speaking** there are two general divisions of the subject:

1. Hydrostatics;
2. Hydrodynamics

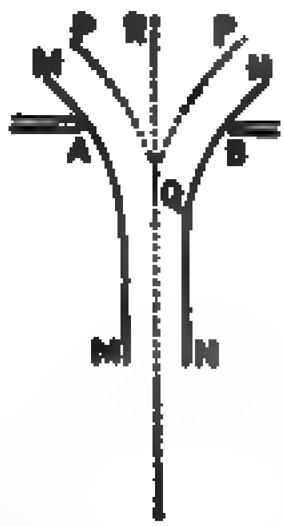


FIGS. 1,198 and 1,199.—*Hydraulic principles: 1. Fluids rise to the same level in the opposite arms of a U tube.* Let A B C, be a recurved tube; if water be poured into one arm of the tube it will rise to the same height in the other arm because the pressure acting upon the lowest part at B, in opposite directions, is proportioned to its depth below the surface of the fluid. Therefore, these depths must be equal, that is, the height of the two columns must be equal, in order that the fluid at B, may be at rest. Unless this part be at rest, the other parts of the column cannot be at rest. Moreover, since the equilibrium depends on nothing else than the heights of the respective columns, therefore, the opposite columns may differ to any degree in quantity, shape, or inclination to the horizon. Thus, if vessels and tubes vary diversely in shape and capacity, as in fig. 1,199, be connected with a reservoir, and water be poured into any one of them, it will rise to the same level in all of them. The reason of this fact will be further understood from the application of the principle of equal moments, for it will be seen that the velocity of the columns, when in motion, will be as much greater in the smaller than in the larger columns, as the quantity of matter is less; and hence the opposite moments will be constantly equal. Hence, water conveyed in aqueducts or running in natural channels, will rise just as high as its source. Between the place where the water of an aqueduct is delivered and the spring, the ground may rise into hills and descend into valleys, and the pipes which convey the water may follow all the undulations of the country, and the water will run freely, provided no pipe be laid higher than the spring.

Hydrostatics refers to liquids ***at rest***, and hydrodynamics to liquids ***in motion***. The outline here given relates to water.

Water.—Those who have had experience in the design or operation of pumps, have found that water is an unyielding substance when confined in pipes and pump passages, thus necessitating very substantial construction to withstand the pressure, and periodic shocks or water hammer.

Ques. What is the most remarkable characteristic of water?



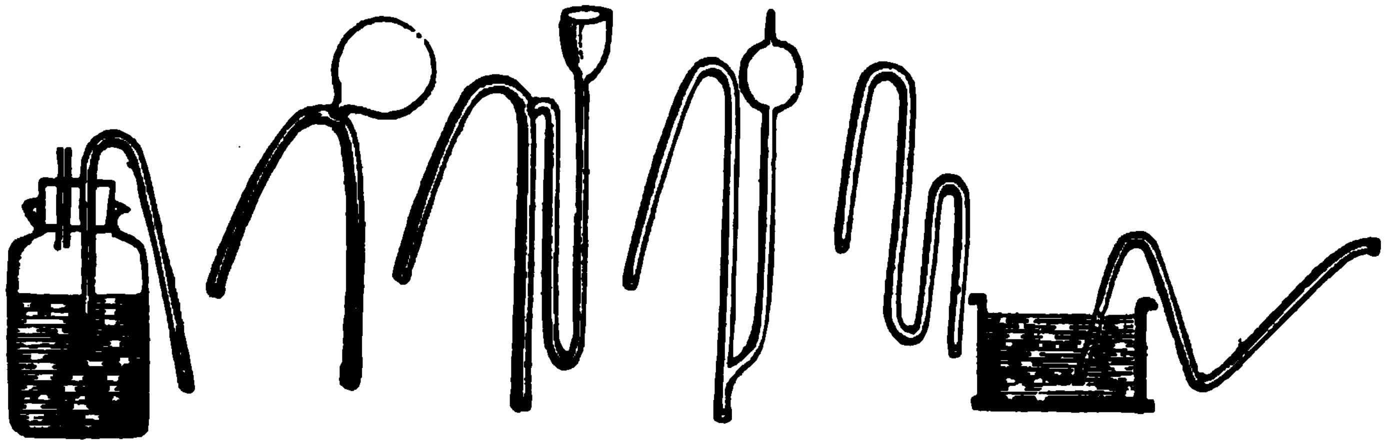
Figs. 1,200 to 1,202.—***Flow of Water Under Pressure.*** If an orifice in a vessel look downward and the column of liquid over it be short, this will simply drop out by its own weight, starting at a velocity of 0. But if a considerable depth of liquid be above, its gravity produces a corresponding pressure on its base, or on that liquid which is near it; so that, if a plug be removed from an orifice in or close to the base, the liquid starts at once into rapid motion. Each particle of a jet A issuing from the side of a vessel Fig. 1,200, moves horizontally with the velocity above mentioned, ***but it is at once drawn downward by the force of gravity***, describing a ***parabola*** in the same manner as a bullet fired from a gun, with its axis horizontal. If the water issue through orifices which are small in comparison with the contents of the vessel, the jets from orifices at different depths below the surface take different forms, as shown at D. ***Quantity of Efflux.***—If the bottom of a vessel containing water be thin, and the orifice be a small circle whose area is A (see fig. 1,201) where AB represents an orifice in the bottom of a vessel, every particle above AB tries to pass out of the vessel, at once, and in so doing exerts a pressure on those nearest. Those that issue near A and B, exert pressures in the directions MM and NN; those near the center of the orifice in the direction RQ, those in the intermediate parts in the directions PQ, PQ. In consequence, the water within the space PQP is unable to escape, and that which does escape, instead of assuming a cylindrical form, at first contracts, and takes the form of a truncated cone. It is found that the escaping jet continues to contract until at a distance from the orifice about equal to the diameter of the orifice; this part of the jet is called the ***vena contracta*** or contracted vein. ***Influence of tubes on the quantity of efflux.***—If a cylindrical or conical efflux tube is fitted to the aperture, the amount of the flow is considerably increased. A short tube, whose length is from two to three times its diameter, has been found to increase the actual efflux per second to about 82 per cent of the theoretical. In this case, the water, on entering the tube, forms a contracted vein, fig. 1,202, just as it would do on issuing freely into the air, but afterwards it expands, and because of the adhesion of the water to the interior surface of the tube, has, on leaving the tube, a section greater than that of the contracted vein. The contraction of the jet within the tube shown in block in the figure, causes a partial vacuum.

Ans. Water at its maximum density (39.1° Fahr.) will expand as heat is added, and it will also expand slightly as the temperature falls from this point.

For ordinary calculations the weight of 1 cu. ft. of water is taken at 62.4 lbs., which is correct when its temperature is 53° Fahr.

The figure 62.5 usually given is approximate. The highest authoritative figure is 62.428. At 62° Fahr., the figures range from 62.291 to 62.36. The figure 62.355 is generally accepted as the most accurate.

The weight of a U. S. gallon of water, or 231 cu. ins. is roughly 8⅓ lbs.



FIGS. 1,203 to 1,208.—*Hydraulic Principles: 2, The siphon.* This device consists of a bent pipe or tube with legs of unequal length, used for drawing liquid out of a vessel by causing it to rise in the tube over the rim or top. For this purpose the shorter leg is inserted in the liquid, and the air is exhausted by being drawn through the longer leg. The liquid then rises by the pressure of the atmosphere and fills the tube and the flow begins from the lower end. The general method of use is to fill the tube in the first place with the liquid, and then, stopping the mouth of the longer leg, to insert the shorter leg in the vessel; upon removal of the stop, the liquid will immediately begin to run. The flow depends upon the difference in vertical height of the two columns of the liquids, measured respectively from the bend of the tube, to the level of the water in the vessel and to the open end of the tube. The flow ceases as soon as, by the lowering of the level in the vessel, these columns become of equal height or when this level descends to the end of the shorter leg. The atmospheric pressure is essential to the support of the column of liquid from the vessel up to the top of the bend of the tube, and this height is consequently limited; at sea height the maximum height is a little less than 34 feet for water, but this varies according to *the density of the fluid.*

Head and Pressure.—These are the two primary considerations in hydraulics. The word head signifies *the difference in level of water between two points*, and it is usually expressed in feet. There are two kinds of head:

1. Static head;
2. Dynamic head.

The static head *is the height from a given point of a column, or body of water* ***at rest,*** considered as causing or measuring pressure.

The dynamic head *is an equivalent or virtual head of water in motion which represents the resultant pressure due to the height of the water from a given point, and the resistance to flow due to friction.*

Thus, when water is made to flow through pipes or nozzles there is a loss of head. These terms are illustrated in fig. 1,209. Here the dynamic head is *greater* than the static head in the supply line to the tank, and *less* in the tank discharge line because of frictional resistance to the flow

FIG. 1,209.—View of elevated tank with pump in operation maintaining the supply which is being drawn upon as shown, illustrating the terms static lift, dynamic lift, static head, and dynamic head.

of the water. In ordinary calculations, it is common practice to estimate that every foot head is equal to one-half pound pressure per sq. in., as this allows for ordinary friction in pipes.

Ques. In pump operation what is the total static head?

Ans. The static lift plus the static head.

Ques. What is the total dynamic head?

Ans. The dynamic lift plus the dynamic head.

Lift.—When the barometer reads 30 inches at sea level, the pressure of the atmosphere at that elevation is 14.74 lbs. per sq. in., that is to say, this pressure will maintain or balance a

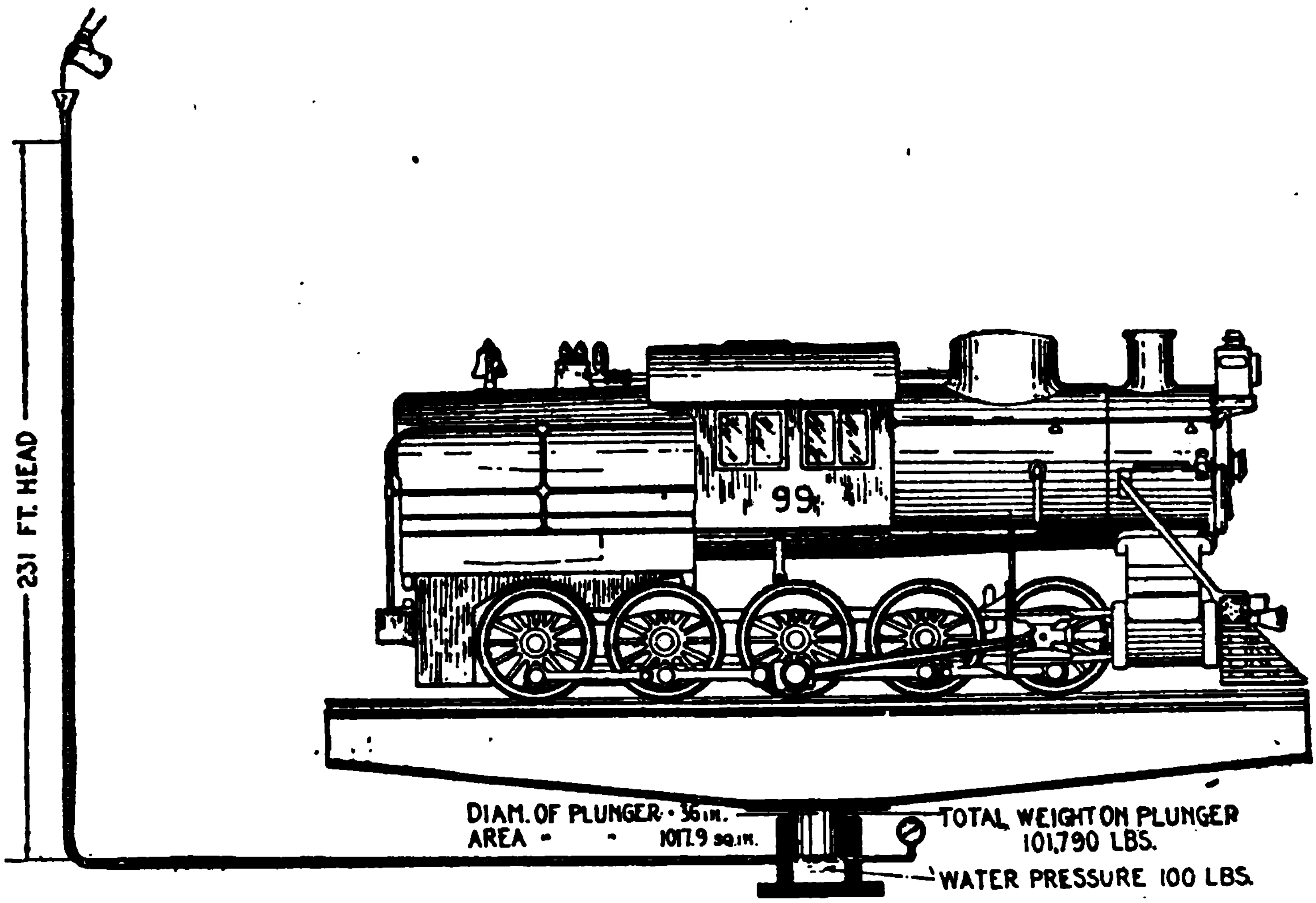


FIG. 1,210.—***Hydraulic principles:*** *3.* Any quantity of water however small may be made to balance any weight however great. The figure shows a locomotive on a turn table balanced by a hydraulic pivot or plunger. Assuming no leakage or friction at the joint, and that the vertical pipe leading to the plunger cylinder is very small, it is evident that it could be filled to the elevation shown with a very small quantity of water—say one quart. If the total weight of locomotive, turn table, etc., and the plunger be 101,709 lbs., and the plunger area be 1,017.9 sq. ins., then the water pressure per sq. in. on the piston necessary to balance the load = 101,796 + 1,017.9 = 100 lbs. Hence the load will be balanced when the pipe is filled with water to a height of 100 × 2.31 = 231 ft.

column of water 34.042 ft. high when the column is completely exhausted of air; and the water is at a temperature of 62° Fahr. In other words, the pressure of the atmosphere then *lifts* the water to such height as will establish equilibrium between the weight of the water and the pressure of the air. Similarly in

pump operation, the receding piston or plunger establishes the vacuum and the pressure of the atmosphere lifts the water from the level of the supply to the level of the pump. Accordingly **lift** as related to pump operation may be defined as ***the height in feet from the surface of the intake supply to the pump.***

Strictly speaking, it is the height to which the water is elevated by atmospheric pressure, which in some pumps may be measured by the elevation of the inlet valves, and in others by the elevation of the piston.

FIG. 1,211.—*Height of a jet.*—If a jet, issuing from an orifice in a vertical direction have the same velocity as a body would have which fell from the surface of the liquid to that orifice, the jet ought to rise to the level of the liquid. It does not, however, reach this; for the particles which fall hinder it. But by inclining the jet at a small angle with the vertical it reaches about nine-tenths of the theoretical height, the difference being due to friction and to the resistance of the air. *The quantities of water which issue from orifices of different areas are very nearly proportional to the size of the orifice,* provided the level remain constant, and this is true irrespective of the form of the opening which may be round, square, or any other shape.

Ques. What is the practical limit of lift?

Ans. 20 to 25 feet.

Long inlet lines, multiplicity of inlet elbows, and high temperature of the water require shorter lifts.

NOTE.—*Flow of water in pipes.* The quantity of water discharged through a pipe depends upon: 1, the *head*, that is, the vertical distance between the level surface of still water in the chamber at the entrance end of the pipe and the level of the center of the discharge end of the pipe; 2, the length of the pipe; 3, the character of its interior surface as to smoothness, and 4, the number and sharpness of the bends, but is independent of the position of the pipe, as horizontal, or inclined upward or downward. The head, instead of being an actual distance between levels, may be caused by pressure, as by a pump, in which case the head is calculated as a vertical distance corresponding to the pressure, 1 lb. per sq. in. = 2.309 ft. head, or 1 ft. head = .433 lb. per sq. in. The total head operating to cause flow *is divided into three parts:* 1, *the velocity head*, which is the height through which a body must fall in a vacuum acquire the velocity with which the water flows into the pipe $= v^2 \div 2g$, in which v is the velocity in ft. per sec. and $2g = 64.32$; 2, *the entry head* required to overcome the resistance to entrance to the pipe. With sharp edged entrance, the *entry head = about ½ the velocity head;* with smooth rounded entrance, the entry head is inappreciable; 3, *the friction head*, due to the frictional resistance to flow within the pipe. In ordinary cases of pipes of considerable length the sum of the entry and velocity heads required scarcely exceeds one foot. In the case of long pipes with low heads, the sum of the velocity and entry heads is generally so small that it may be neglected.

Ques. Why must the lift be reduced as the temperature of the water is increased?

Ans. Because the boiling point of water corresponds to the pressure.

Theoretically a perfect pump will draw water from a height of 34 ft. when the barometer reads 30 ins., but since a perfect vacuum cannot be obtained on account of valve leakage, air contained in the water *and the vapor of the water itself,* the actual height is generally less than 30 feet, and for warm or hot water considerably less.

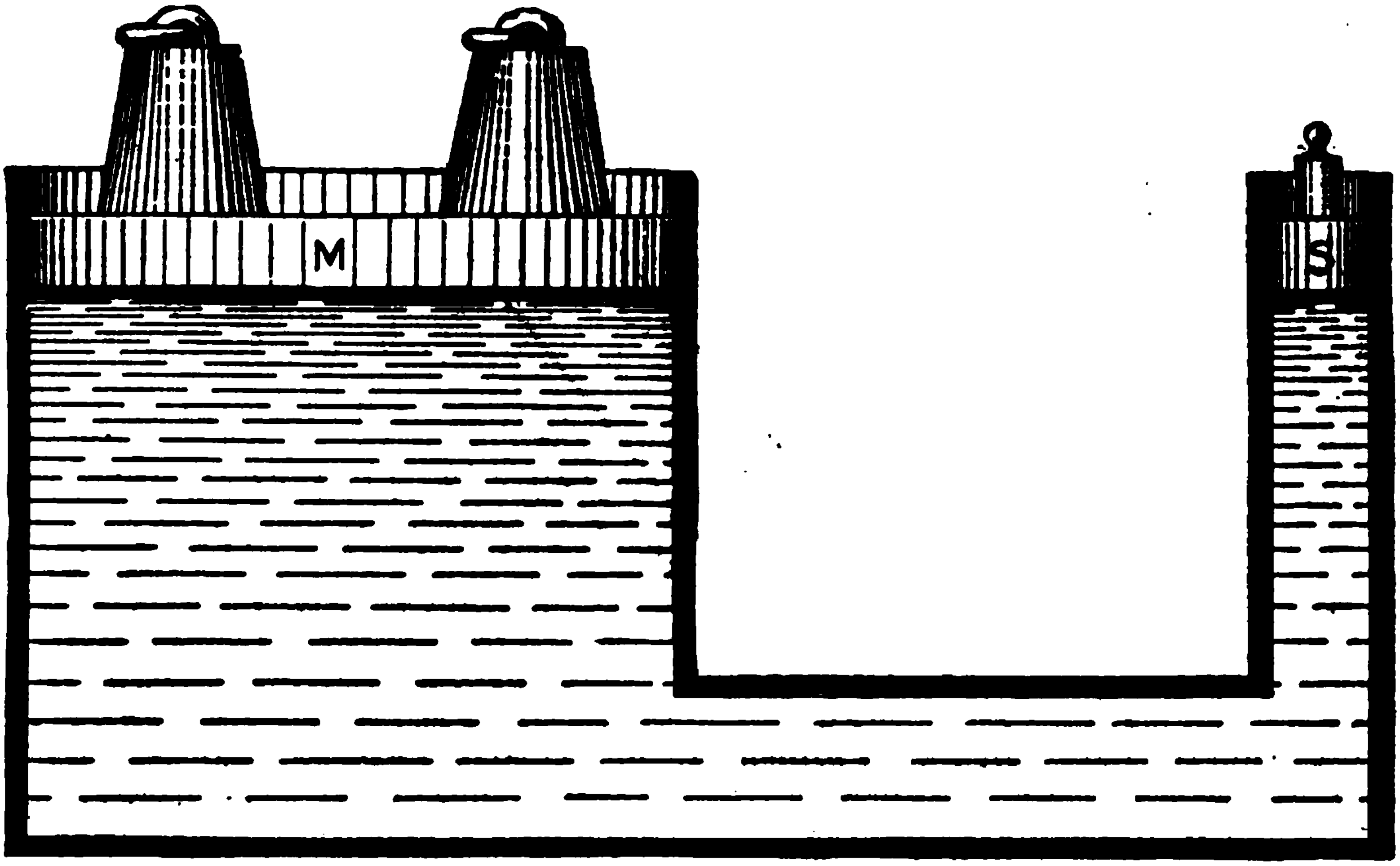


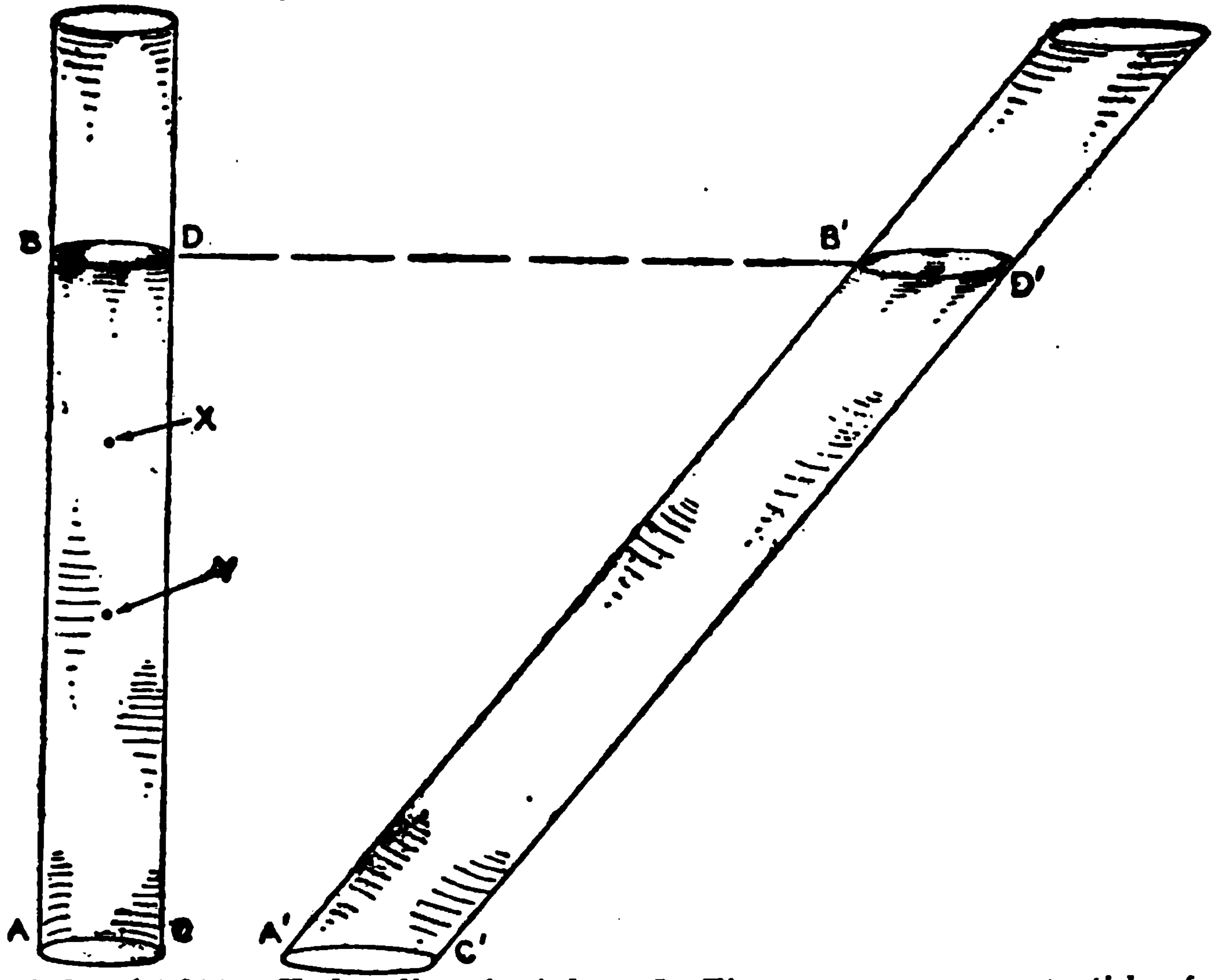
FIG. 1,212.—***Hydraulic principles: 4. The pressure exerted by a liquid on a surface is proportional to the area of the surface.*** Two cylinders of different diameter are joined by a tube and filled with water. On the surface of the two pistons M and S, which hermetically close the cylinders, but move without friction. Let the area of the large piston M be, say thirty times that of the smaller one S, and let a weight, say of two pounds, be placed upon the small piston. The pressure will be transmitted to the water and to the large piston, and as this pressure amounts to two pounds in each portion of its surface equal to that of the small piston, the large piston must be exposed to an upward pressure thirty times as much, or 60 lbs. If now a 60 lb. weight be placed upon the large piston, both pistons will remain in equilibrium, but if the weight be greater or less, the equilibrium will be destroyed.

When the water is warm, the height to which it can be lifted decreases, on account of the increased pressure of the vapor. That is to say, for illustration, a boiler feed pump taking water at say 153° Fahr., could not produce a vacuum greater than 21.78 ins., because at that point

the water would begin to boil and fill the pump chamber with steam. Accordingly, the theoretical lift corresponding would be

$$34\times\frac{21.78}{30}=24.68 \text{ ft. approximately.}$$

The result is approximate because no correction has been made for the 34 which represents a 34 foot column of water at 62°; of course, at



FIGS. 1,213 and 1,214.—***Hydraulic principles:*** *5. The pressure upon any particle of a fluid of uniform density is proportional to its depth below the surface.* ***Example 1.*** Let the column of fluid ABCD, fig. 1,213, be perpendicular to the horizon. Take any points, X and Y, at different depths, and conceive the column to be divided into a number of equal space by horizontal planes. Then, since the density of the fluid is uniform throughout, the pressure upon X and Y, respectively, must be in proportion to the number of equal space above them, and consequently in proportion to their depths. ***Example 2.*** Let the column be of the same perpendicular height as before, but inclined as is fig. 1,214; then its quantity, and of course its weight, is *increased* in the same ratio as its length exceeds its height; but since the column is partly supported by the plane, like any other heavy body, the force of gravity acting upon it is *diminished* on this account in the same ratio as its length exceeds its height; therefore as much as the pressure on the base would be augmented by the increased length of the column, just so much it is lessened by the action of the inclined plane, and the pressure of any part of C'D' will be, as before, proportioned to its perpendicular depth, and the pressure of the inclined column A'C'D'B' will be the same as that of the perpendicular column ABCD.

153° the length of such column would be slightly increased.

It should be noted that the figure 24.68 ft. is the *approximate* theoretical lift for water at 153°; the *practical* lift would be considerably less.

The following table shows the theoretical maximum lift for different temperatures; leakage not considered.

Theoretical Lift for Various Temperatures

Temp. Fahr.	Absolute pressure of vapor lbs. per sq. ins.	Vacuum in inches of mercury	Lift in feet	Temp. Fahr.	Absolute pressure of vapor lbs. per sq. ins.	Vacuum in inches of mercury	Lift in feet
102.1	1	27.88	31.6	182.9	8	13.63	15.4
126.3	2	25.85	29.3	188.3	9	11.6	13.1
141.6	3	23.83	27	193.2	10	9.56	10.8
153.1	4	21.78	24.7	197.8	11	7.52	8.5
162.3	5	19.74	22.3	202	12	5.49	6.2
170.1	6	17.70	20	205.9	13	3.45	3.9
176.9	7	15.67	17.7	209.6	14	1.41	1.6

Elementary Pumps.—There are three elements necessary for the operation of a pump:

1. Inlet or suction valve;
2. Piston or plunger;
3. Discharge valve.

Simple pumps may be divided into two classes:

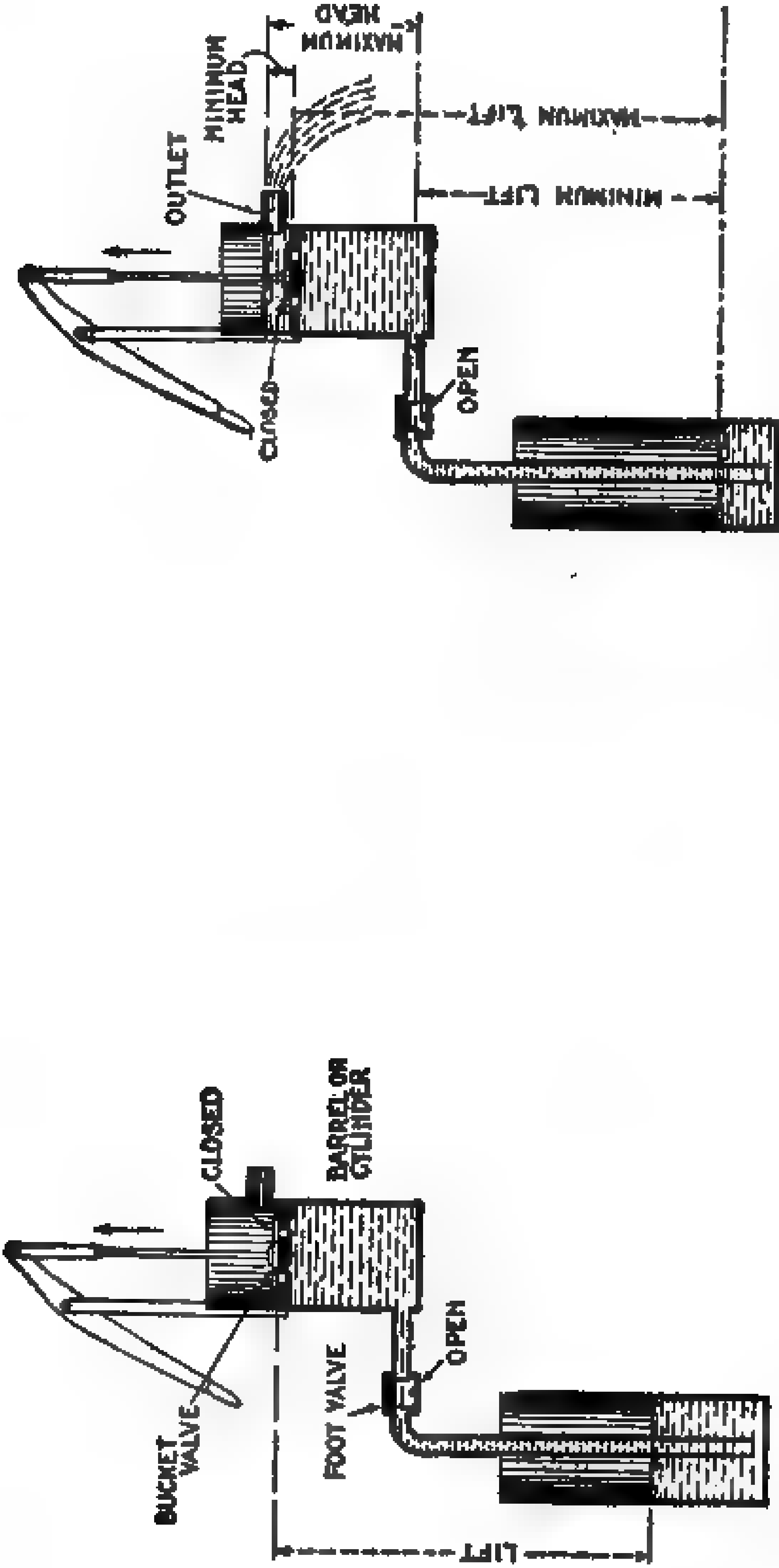
1. Lift pumps;
2. Force pumps.

A lift pump is one which does not elevate the water higher than the lift; a force pump operates against both lift and head.

Lift Pumps.—Figs. 1,215 to 1,217 show the essentials and working principle of a simple lift pump.

In construction there are two valves in this type of pump, which are known as the foot valve and the bucket valve. *In operation* during the up stroke the bucket valve is closed and foot valve open, allowing the atmosphere to force the water into the cylinder.

When the piston begins to descend, the foot valve closes and bucket



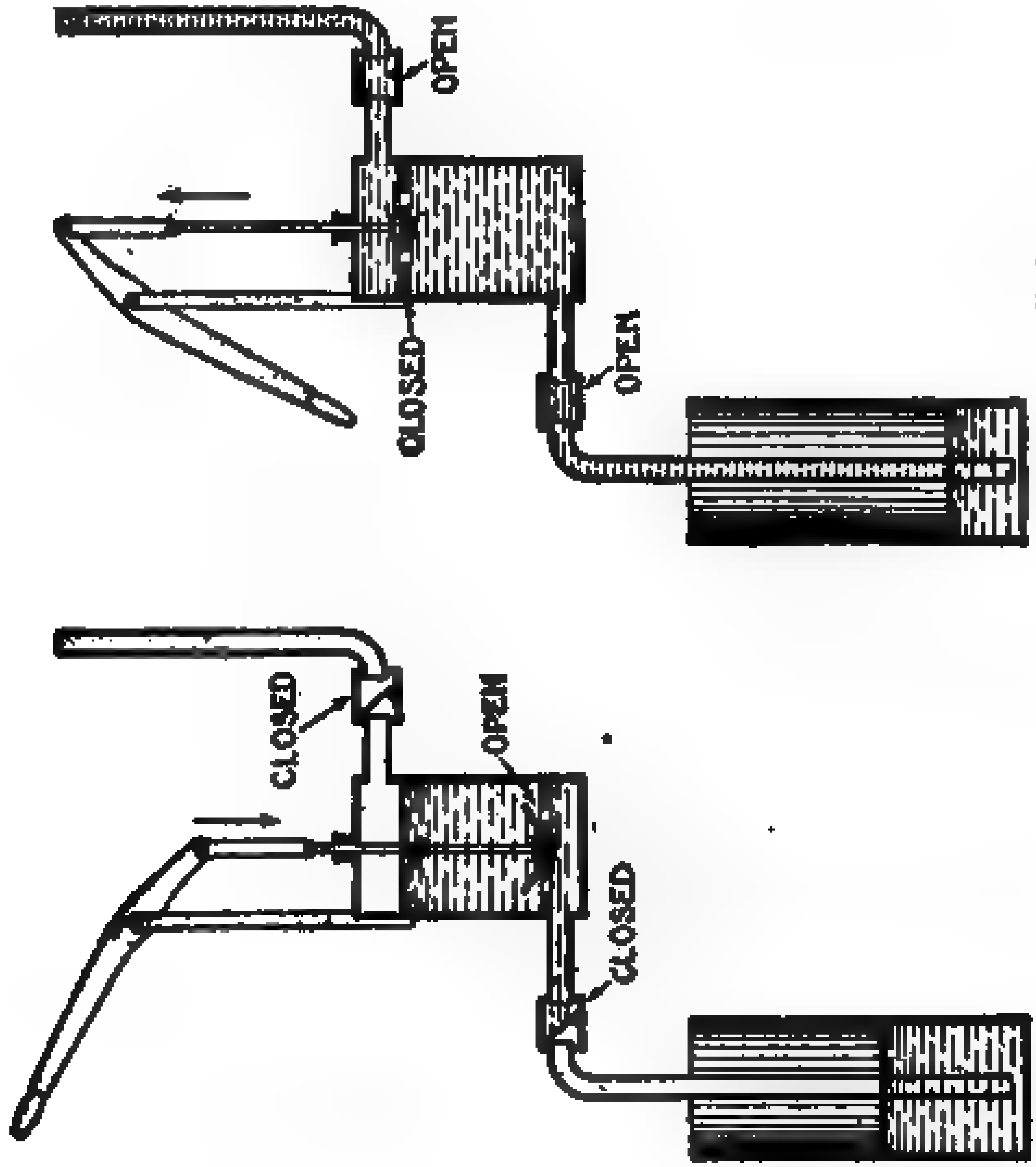
FIGS. 1,216 to 1,217.—Elementary single acting lift pump showing essential features and cycle of operation.

valve opens, which transfers the water in the cylinder from the lower side of the piston to the upper side as in fig. 1,216.

During the next up stroke, the water, already transferred to the upper side of the piston, is discharged through the outlet as in fig. 1,217.

It will be noted that as the piston begins the up stroke of discharge it is subject to a small maximum head, and at the end of the up stroke to a minimum head as indicated in fig. 1,217. This variable head is so small in comparison to the head against which a force pump works that it is not ordinarily considered.

Force pumps.—The essential feature of a force



FIGS. 1,219 to 1,220.—Elementary single acting force pump showing distinguishing feature of closed cylinder.

pump which distinguishes it from a lift pump is that *the cylinder is always closed*, whereas in a lift pump it *is* ***alternately*** *closed and open* when the piston is respectively at the upper and lower ends of its stroke.

As shown in figs. 1,218 to 1,220, the cylinder top is closed by a cover, the piston rod passing through a stuffing box; this keeps the cylinder closed

In addition to the foot and bucket valves of the lift pump, a head valve is provided.

FIGS. 1,221 ... parts. The
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In operation, during the up stroke, atmospheric pressure forces water into the cylinder as in fig. 1,218; during the down stroke this water is transferred from the lower to the upper side of the piston as in fig. 1,219; during the next up stroke, the piston forces the water out of the cylinder through the head valve which closes when the piston reaches the end of the stroke and the cycle is repeated. The positions of the valve are shown in the cuts.

A simple form of force pump, is one known as a single acting plunger pump, a type extensively used, its cycle of operation being shown in figs. 1,221 and 1,222. The figures show the distinguishing features, such as closed cylinder, plunger, and only two valves.

In operation during the up stroke water fills the cylinder, inlet valve opens, and outlet valve closes, as shown in fig. 1,221 During the down stroke, the plunger "displaces" the water in the barrel, forcing it through the discharge valve against the pressure due to the head.

Ques. What is the difference between a piston and a plunger?

Ans. A piston is *shorter* than the stroke, whereas a plunger is *longer* than the stroke.

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Double Acting Force Pump.—*By fitting a set of inlet and* ***outlet*** *valve at each end of a pump cylinder it is rendered* **double acting,** that is, a cylinder full of water is pumped each stroke instead of every other stroke.

With this arrangement the piston need have approximately only half the area of the single acting piston for equal displacement, and accordingly the maximum stresses brought on the reciprocating parts are reduced approximately one-half, thus permitting lighter and more compact construction.

In the double acting pump there are no bucket valves, a solid piston being used. The essential features and operation are plainly shown in figs. 4,259 and 4,260. There are two inlet valves A, B, and two discharge valves C, D, the cylinder being closed and provided with a piston.

In operation, during the down stroke, water follows the upper face of the piston through valve A. At the same time the previous charge is forced out of the cylinder through valve D, by the lower face of the piston. During these simultaneous operations, valves A and D, remain open, and B and C, closed, as in fig. 4,259.

During the up stroke, water follows the lower face of the piston through valve B. At the same time, the previous charge is forced out of the cylinder through valve C, by the upper face of the piston. During these simultaneous operations, valves B, and C, remain open, and A, and D, closed.

Classification of Pumping Engines.—The numerous and varied conditions of service have resulted in the manufacture of several types of pumping engine of widely different construction.

All pumps may be divided into two classes: 1. *Reciprocating;* 2. *rotary;* and with respect to service, construction, etc., they may be classified in several ways:

1. With respect to the cycle of operation at the water end, as

a. Single acting;
b. Double acting.

2. With respect to the construction of the water end, as

a. Plunger;
b. Piston;
c. Inside packed;
d. Outside packed.

3. With respect to the steam features, as

a. Compound;
b. Triple expansion;
c. Quadruple expansion.

4. With respect to general construction, as

a. Single;
b. Duplex;
c. Horizontal;
d. Vertical;
e. Direct acting;
f. Fly wheel { direct connected / with walking beam.

Conditions of Operation.—Water, as before stated, is an unyielding mass, and its behavior in passing through the water end of the pump is quite different from that of the steam at the other end. Accordingly severe strains in the nature of shocks are brought on the parts through which the water traverses, and the construction must be very substantial to resist this action.

Water, being a heavy body, has considerable inertia; this, together with the intermittent action of the pump necessarily limits the speed of the piston or plunger, resulting in a machine of considerable size in comparison with a steam engine of equal horse power.

In practice, the speed of a pump is only about one-quarter that of a steam engine, hence large steam cylinders are required for a given capacity at the water end. An excess of surface, then, is exposed to radiation, and owing to the slow piston movement, and large size, there is more chance for leakage of steam. These are not conditions conducive to economy, yet the pumping engine has at times held world records for economical steam consumption. The reason for this is because, in many installations, the load is constant, whereas with other classes of engines, the best results are not obtainable, owing to the necessity of proportioning the engine to meet the requirements of a variable load.

On account of the constant load the cylinders of a pumping engine may be so proportioned, and the steam distribution adjusted, that the maximum economy that can be obtained within the limits of the working pressure is secured.*

*NOTE.—The importance of properly proportioning an engine to the work it has to do is emphasized by the high economy of the pumping engine notwithstanding its slow piston speed. According to tests of Profs. Deuton and Jacobus on a 17×30 fixed cut off non-condensing engine, the loss in economy for about one-fourth cut off is at the rate of one-twelfth lb. of water per horse power for each decrease of a revolution per minute from 86 to 26 revolutions, and at the rate of five-eighths lb. of water below 26 revolutions

Reciprocating Pumps.—The large variety of pump represented under this heading are used for almost every condition of service. They are either single or double acting, single or multi-cylinder, vertical or horizontal, piston or plunger, etc., as may be best suited to any particular condition of service.

The principles of operation have been given under elementary pumps, and the accompanying cuts illustrate the trend of design and construction.

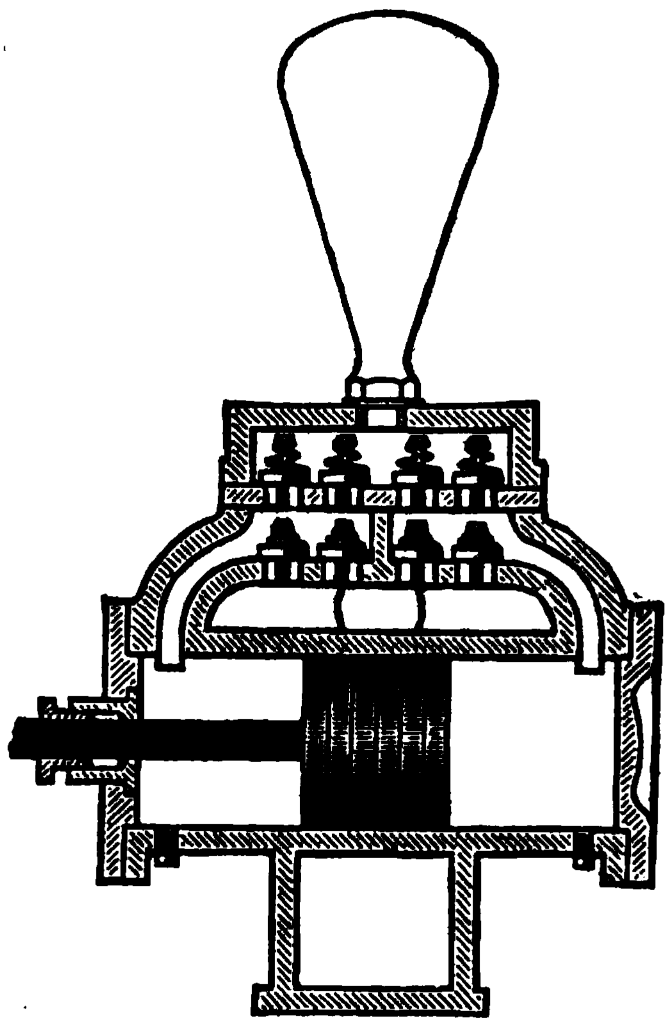


FIG. 1,225.—Double acting piston water end, showing sectional view of piston, cylinder, stuffing box, valves, and water passages. The lower row of valve are the inlet valves, and the upper row the discharge valves.

Water Ends.—There are, properly speaking, four kinds of water end to power pumps:

1. A piston packed with fibrous material within the cylinder, as shown in fig. 1,225. The letter P in fig. 1,226 and the following cuts indicates the plunger.

2. Inside packed plunger, with a stuffing box used for heavy pressures in hydraulic apparatus, or as shown in fig. 1,226 for larger plungers.

3. A single acting outside packed plunger as in fig. 1,227.

4. Two plungers, fig. 1,228, connected outside of the cylinder with a stuffing box in two cylinder heads, through which the plungers work.

The construction of the water ends of single cylinder and duplex pumps is practically the same; any slight differences which may be found are confined to minor details which in no way affect the general design or operation of the pump.

Pump Valves.—The valve apparatus is perhaps the most important part of any form of pump and its design has a material bearing upon its efficiency.

FIG. 1,226.—Double acting inside packed plunger water end showing sectional view of working parts.

The valves shown in fig. 1,225 are carried by two plates or decks, the suction valves being attached to the lower plate and the delivery valves to the upper one. The upper deck, and sometimes both decks, are removable.

The valves are secured to the seats by means of bolts or long screws, which, in turn, are screwed into the seat, as shown in figs. 1,229 and 1,231 or capped as in fig. 1,232.

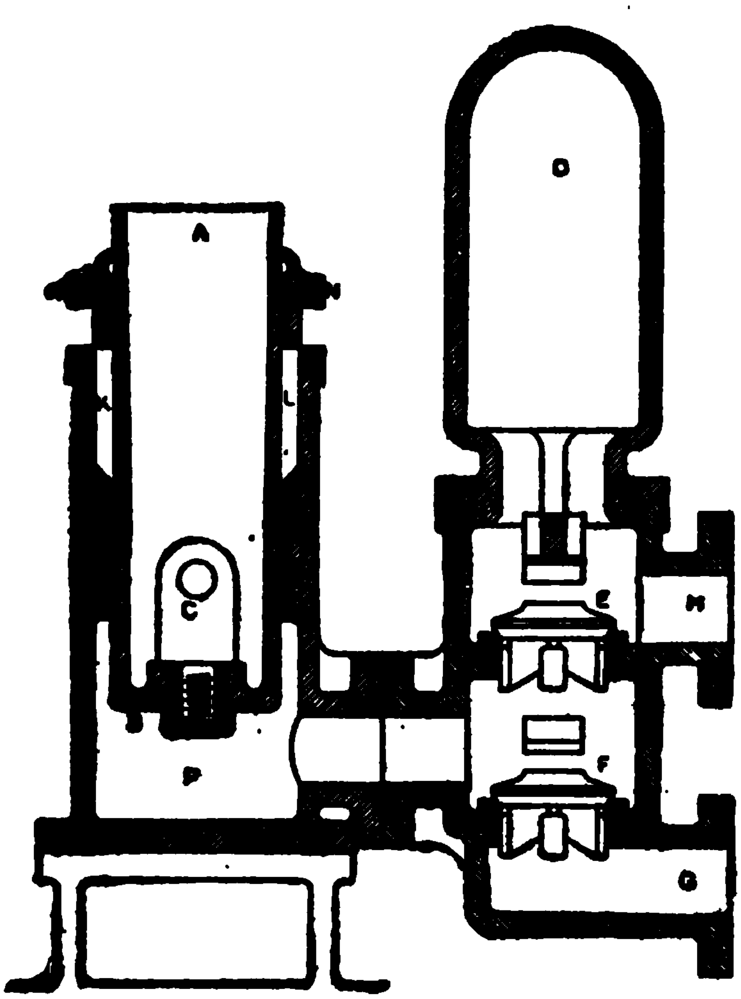


FIG. 1,227.—**Single acting outside packed plunger pump.** ***In construction,*** **the moving part consists of the plunger AB working in the stuffing box KL. There are two valves or sets of valve, F and E. The stuffing box KL being on the outside can be kept in perfect adjustment. and with proper design the suction and discharge valves may be examined by the simple removal of a bonnet. The strong points of this pump are its simplicity, and the ready accessibility for examination and adjustment of all parts on which the operation of the pump may depend.**

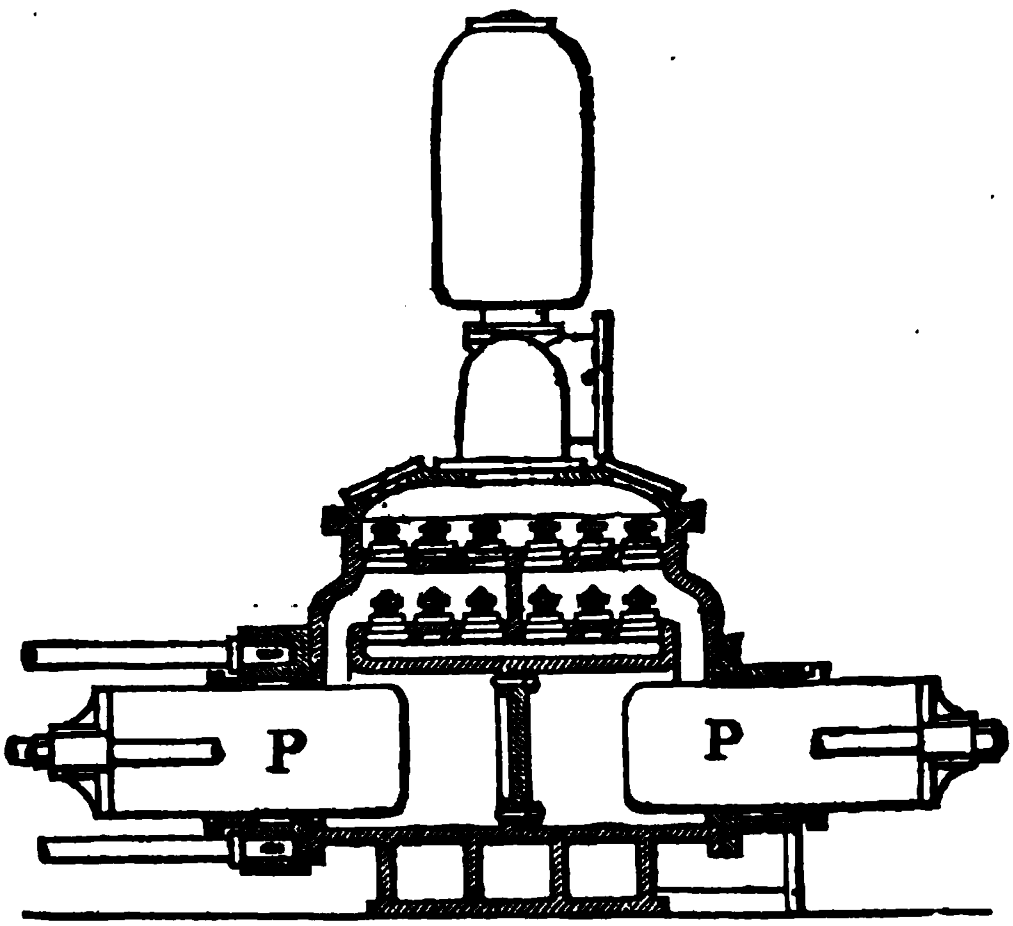


FIG. 1,228.—Double acting outside packed plunger water end. These plungers are connected by yokes and outside rods, the yokes and a portion of the rods being shown in the figure. The construction is virtually a combination of two single acting plunger pumps so connected as to give the equivalent of a double acting pump cycle.

The valves in all pumps except the large sizes, which may properly be classed with pumping engines, are of *the flat rubber disc type*, with a hole in the center to enable the valve to rise easily on the bolt, the latter serving as a guide.

A conical spring is employed to hold the valve firmly to its seat, the spring being held in position by the head of the bolt, or cap, as shown.

Certain improvements in pump valves have been made which tend to increase the durability and to prevent the liability of sticking, which is not an uncommon occurrence after the valves have become badly worn. The improved forms of pump valve are shown in figs. 1,231 and 1,232. When these valves leak, through wear, the disc may be reversed, using the upper side of the disc next to the valve seat. This

Figs. 1,229 to 1,232.—Various details of pump valve construction.

can be done with ordinary valves also, provided the spring has not injured the upper surface of the disc.

Valve seats are generally pressed into the plates, although instances may be found where they are screwed. When pressed in they may be withdrawn by substituting a bolt having longer length of screw thread than the regular bolt, and provided with a nut and yoke, as shown in fig. 1,233. The bolt is slipped through a yoke and screwed into the seat. By turning the nut the seat can generally be started without difficulty.

Fig. 1,234 represents the customary *gland and stuffing box*, in which the gland is adjusted by the nuts C and D upon two studs. After the

adjustment has been properly made, lock nuts are tightened which leaves the gland free, yet preserves the alignment.

It has been proven by practice, after long and costly experiments, that a number of small valves instead of one large valve are more durable. Worthington, Dunham, Leavitt, Holly and other leading pump engineers had occasion to find the truth of this statement early in their careers.

H. F. Dunham confines his practice to four, or four and one-half inch valves in all cases except for pumps of very small capacity; the author considers this good practice, as larger valves involve too great lift, and the smaller sizes necessitate an undue multiplicity of valve unit. The

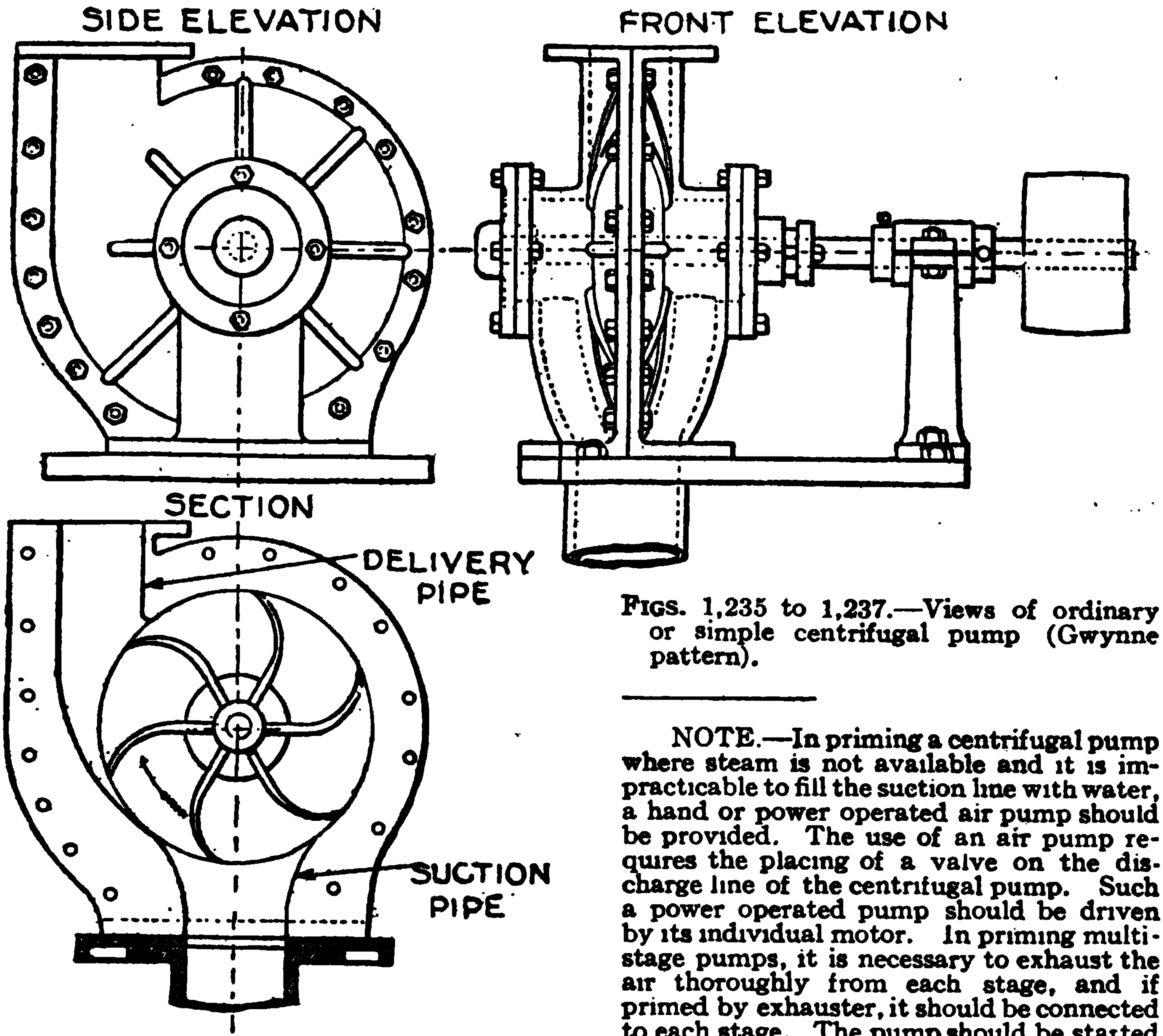
FIG. 1,233.—Jig for removing valve seats of the press fit type. It consists of a bolt with extra length of thread, a yoke and nut as shown. The operation is apparent from the cut.

FIG. 1,234.—Detail of stuffing box for piston pump.

"slamming" of large valves under moderate speeds proved itself a difficulty hard to overcome, until the principle of keeping the valve area as low as possible within reasonable limits had been fully demonstrated.

Rotary Pumps.—This type of pump may be defined, as *one having a revolving piston, or pistons which partake of the nature of cams, rotating upon an axis and being in contact at one or more points with the walls of the enclosing chamber.* In operation, a rotary pump continuously "scoops" the water from its chamber, the operation being somewhat similar to bailing a boat with a scoop.

Centrifugal Pumps.—This type of pump may be defined as *one in which curved vanes or impellers, rotating inside a close fitting casing, draw in the liquid at the center and, by virtue of*



FIGS. 1,235 to 1,237.—Views of ordinary or simple centrifugal pump (Gwynne pattern).

NOTE.—In priming a centrifugal pump where steam is not available and it is impracticable to fill the suction line with water, a hand or power operated air pump should be provided. The use of an air pump requires the placing of a valve on the discharge line of the centrifugal pump. Such a power operated pump should be driven by its individual motor. In priming multistage pumps, it is necessary to exhaust the air thoroughly from each stage, and if primed by exhauster, it should be connected to each stage. The pump should be started only after it is entirely filled with water. The pump must not be run emply, as the clearance rings and shaft sleeves, which in good designs have very small clearances, will bind, heat and cut if run dry. When first starting the motor, be sure to see that its direction of rotation agrees with that of the pump, as pump must not run in a direction opposite to that for which it is intended. This will be usually stamped on the casing or may be marked on the blue print of the pump. After the pump is primed, the shaft should be turned over one or two revolutions to allow all air to free itself from the vanes of the impeller.

NOTE.—Before starting a centrifugal pump and its motor, care should be taken to clean the bearings, as dirt and substances may get in during shipment or erection. They should then be filled with a pure, clean mineral oil. This oil should be changed when it becomes dirty and the bearings thoroughly cleaned at the same time. At regular intervals these bearings should be examined.

NOTE.—In operating centrifugal pumps, where there is a considerable amount of air or gases in the water, the air stop cock on top of casing should be opened occasionally; in extreme cases, the cock may be left partially open.

centrifugal force, *throw out the liquid through an opening at the periphery of the casing.*

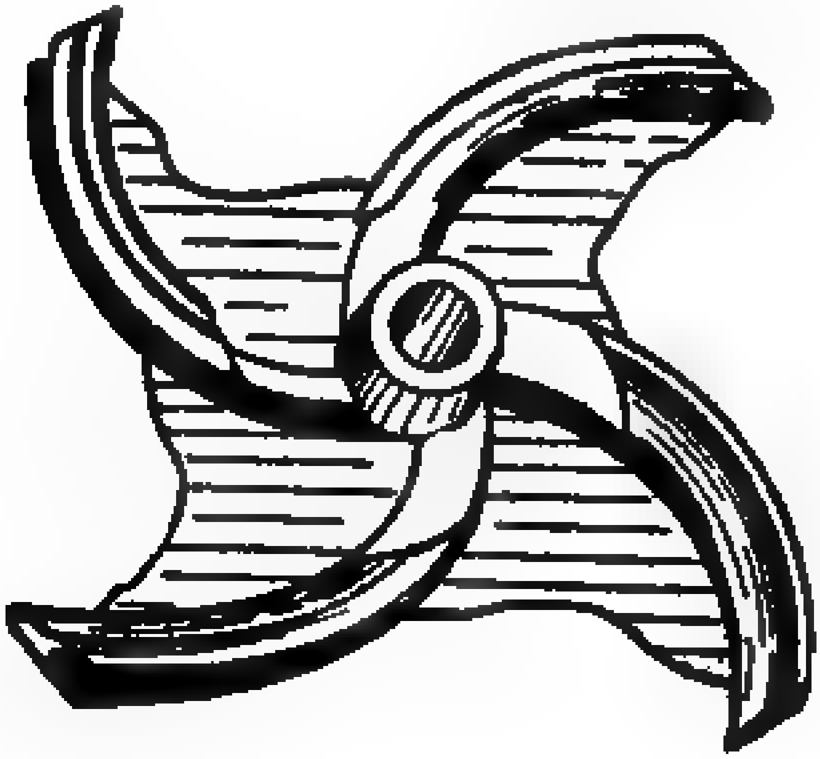
Centrifugal pumps are divided into four classes:

1. Simple;
2. Conoidal;
3. Volute;
4. Turbine { single stage: multi-stage.

The simple or ordinary type consists of a series of blade. which are rigidly fixed on a shaft and enclosed in what is called the whirlpool chamber When the blades are rapidly revolved, the centrifugal force thus created throws the water through the outlet in the casing.

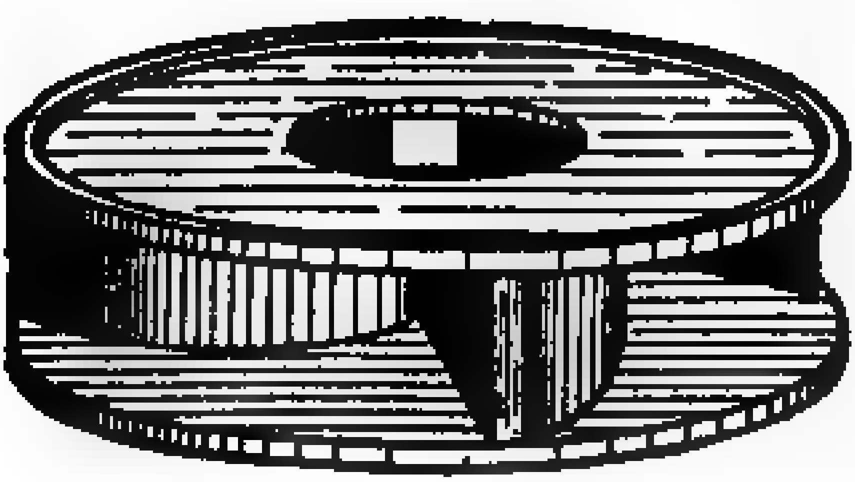
The general appearance of the conoidal pump (named from the cone shaped impeller) is somewhat different from the ordinary centrifugal pump, on account of the widening of the pump chamber to receive a special form of impeller, which consists of a double conical shaped core, on which radial vanes are cast or mounted. The peculiar shape of this

FIGS. 1,238 and 1,239.—Gwynne's conoidal type centrifugal pump and detail of impeller. The arrows indicate the direction in which the impeller turns. ***In construction,*** A, is the inlet; B, discharge; C, the bottom chamber within which the impeller consisting of disc K, and blades or vanes 1 to 8, revolve closely without touching the surrounding casing. To chamber C, are attached the conducting case D and D′, which form between them an easy curved spiral discharge passage, gradually enlarging toward its outlet B. The shaft G, passes through the stuffing box F, thence through and into the sleeve R, which is securely bolted by wide flanges to the case D′, thus forming a central rigid and long bearing for the spindle. The driving pulley P, has its bearing babbitted and runs on the outer diameter of the sleeve R. S is the driving clutch secured to spindle by the feather key and set screw S′, and engages by the lugs to the driving pulley. Means of lubrication is afforded by the oil holes r. The vanes on disc K, extend above the disc to exclude dirt from the bearings, and partially relieve the downward pressure upon the disc. The spaces above and below the disc are connected by the holes *kk* through the disc, equalizing the vacuum therein, relieving it from downward pressure, and balancing it so that no lower bearing is required.



core serves to modify gradually the direction of the incoming current, thereby preventing waste of power. The pump chamber is divided into two parts by a radial partition, which extends entirely around the interior of the chambers and encloses the base of the conoidal impellers. This partition prevents the impingement and consequent disturbance of the two entering columns of water. Conoidal pumps are especially suitable for supplying water to surface condensers, or for irrigation, pumping sewage, or purposes where the liquid pumped is accompanied by sand, mud, silt, etc. They are comparatively inexpensive and the space required by them, relative to the quantity of water delivered, is claimed to be about one half that of a centrifugal pump of the ordinary pattern. They are designed for a maximum head of 30 feet.

Volute pumps are built for medium lifts, but for all capacities. They are desirable for heads up to 70 feet, without necessitating the use of pumps, which are either especially large or very expensive. Volute pumps run at moderate speed.



FIGS. 1,240 to 1,242.—Three styles of impeller for centrifugal pumps. Fig. 1,240. shows a form used for small sizes and for thick liquids; fig. 1,241, is a hollow arm type used in large pumps, and has the advantage that the water is thrown outward without any churning action, and that there are no dead spaces; fig. 1,242 is used for dredges and has the advantage that the sand is prevented grinding between the blades and the casing, yet large openings are free for the passage of sand and mud.

The turbine type may be defined as *a centrifugal pump having* **stationary guides** *or diffusion vanes inside the casing.* The diffusion vanes are placed between the periphery of the impeller and the case which take the place of the usual whirlpool chamber and assist in guiding the water to the outlet without internal shock or commotion.

The very limited head at which it was possible to operate the earlier pumps with economy has been overcome by connecting two or more units upon one shaft and operating them in series, that is, passing the water through each unit in succession, thus the head is divided between the units by a multi-stage operation and by providing a sufficient number of units or stages, they may be operated with heads even exceeding two thousand feet without impairing the economy.

Fig. 1, [illegible] direct connection to moderate speed motors or for belt or gear drive. A single suction opening is provided on the side furthest from the driving power, thus affording the facility for connecting the suction pipe. The casing is in one piece with feet for mounting on the base plate. The side plates are removable, affording access to the internal parts of the pump. There is a male and female joint between the side plates and casing insuring perfect alignment. The bearings are fastened to brackets cast integral with the side plates, there being a male and female joint between the bracket and bearing body. The impeller is of the enclosed type of cast iron or bronze as required. Alberger regular volute pumps are of the same type and general design, except that the respective sizes are of greater diameter, thus permitting a slower speed for a given head making this type particularly suitable for engine drive. Standard and regular volutes can be built for heads up to about 85 feet, provided the available speed of the driving power will permit. Standard volute pumps are built in sizes having discharge openings from 1½ inches up to 72 inches in diameter.

rtical three stage turbine pump show-
e arrangement of impellers A, and
ages B. This pump has the suction
t the top; the discharge leaves the
chamber of the last (lowest) impeller
the circle. The shaft rests in a
ring at the top, and is further held
s formed in the successive sections of
At the bottom it is provided with a
ancing arrangement. Each impeller,
oins the guide passages of the pre-
e section, is fitted into the case so as
tight a joint as possible without intro-
great frictional resistance to rotation.
exception of the entrance opening, the
rface of the impeller is exposed to the
essure, so that there is a resultant up-
are on each impeller, equal to the area of its entrance multiplied by the difference between the entrance and discharge pressures of that stage. If all the impellers be alike, the total upward thrust is equal to the product of entrance area multiplied by the total head on the pump. The pumps are so proportioned that this upward thrust slightly exceeds the weight of the rotating portion, consisting of impellers and shaft. *The excess of upward pressure, however, is relieved by the balancing device located at the lower end of the shaft, with the result that the rotating part is precisely balanced, thus relieving the thrust bearing of all load while the pump is running.* The balancing device referred to consists of two chambers, C and D, formed centrally in the bottom of the lowest section of the pump case. The large chamber C encloses a projecting hub E on the lower surface of the impeller. This hub rotates

with the impeller, and the joint between the hub and the walls of the chamber is, therefore, loose enough to allow water from the delivery side of the last impeller to leak into chamber C, and establish the full
sure in that chamber. The small lower
ontains a plug H, which may be ad-
s by means of screws. The forward end
ts closely into a recess in the face of the
recess communicates, by way of the
part of the hub and the passage *g*, with
entrance side of the last impeller. ***In operation,*** when chamber C becomes filled with water, or rather when leakage through the joint around the tube E has raised the pressure in the chamber C to the delivery pressure, the total upward pressure on the impellers is greater than the total weight of the rotating part of the pump. The rotating element is therefore lifted until the recess in hub E is raised clear of the plug H. In this position the pressure in chamber C is relieved through the passage *g*, with the result that the rotating element again settles down over the adjusting plug H. As this action tends to recur, a position of equilibrium is established near the point where the plug just enters the recess in the hub E. The precise position of this point may be altered by the adjusting screws of the plug H, thereby adjusting the endwise position of the impellers in the casing. When the pump is not in operation, of course the upward pressure of the water does not act, and the weight of the rotating part must be carried by the thrust bearing.

Slip of Pump.—This is properly defined as *the difference between the theoretical and the actual discharge of a pump.* It is expressed as a percentage of the theoretical discharge or *displacement.* Slip is due to three causes:

FIGS. 1,245 and 1,246.—Sectional view of water end, *illustrating lift of valve,* and *slip.* Fig. 1,245, lift for slow or moderate speed; fig. 1,246, lift for high speed. The cuts clearly show the relative position of the suction and discharge valves during the movements of the piston. *Pump slip* or slippage is a term used to denote *the difference between the calculated and the actual discharge of a pump,* and is generally expressed as a percentage of the calculated discharge. Thus, when the slippage is given as 15 per cent. it indicates that the loss due to slip amounts to 15 per cent. of the calculated discharge. Slippage is due to two causes, the time required for the suction and discharge valves to seat. When pumps run very fast the piston speed is so high that the water cannot enter the pump fast enough to completely fill the cylinder and consequently a partial cylinder full of water is delivered at each stroke. *High speeds also increase slippage,* due to the *seating of the valves.* Fig. 1,246 represents a sectional view of the water end of a pump, showing the position of the valves during a quick reversal in the direction of the arrows, which illustrates the position of the valves corresponding to high speeds. The valves in a pump, like almost every other part in the operation of machinery, do not act instantaneously, but require time to reach the seats. When pumps run at high speed the piston will move a considerable distance, while the valves are descending to their seats, and water flows back into the pump cylinder until the valves are tightly closed. The valves will remain in the raised position shown in fig. 1,246 until the piston stops at the end of the stroke, and under high speed the piston will reach the position on the return stroke indicated by the dotted line L, by the time the valves are closed. The cylinder will be filled up to this point with water from the delivery chamber so that no vacuum can be formed until after the piston reaches this position. The volume of water that can be drawn into the cylinder must necessarily be represented by the cubic inches of space, minus the quantity which flows back during the time the valves are closing. It will thus be seen that the actual volume of water discharged is considerably less than a cylinderful, and the difference, whatever it may prove to be, is called, and is due to slippage. Fig. 1,245 represents the same pump running at a comparatively low speed. It will be noticed that the valves have not been raised as high as in fig. 1,246, because a longer time being allowed for the discharge of the water, a smaller orifice is sufficient. It will be seen also that the piston, moving at a lower velocity, cannot travel as far in fig. 1,245 before the valves seat, and consequently a vacuum can be created in the cylinder earlier in the stroke, and a larger volume of water can therefore be drawn in during the return stroke. In the latter case it is evident that the volume of water drawn into the cylinder will be nearly equal to a cylinderful and consequently the loss by slippage must be correspondingly less. In order to reduce the loss by slippage several valves are used instead of a single valve of equal area. A flat disc valve will rise a distance equal to one-fourth the diameter of the port or of the opening in the seat to discharge the same volume of water that can flow through the port in the same time. In practice the rise exceeds this proportion of one-fourth a trifle, owing to the friction of the water, and this is especially true at high speeds.

1. Leakage of water past the piston or plunger;
2. Leakage at the valves;
3. Back flow through the valves while closing;
4. Leakage through the stuffing box.

The piston leakage is due usually to poor packing and lack of attention. In water works plants, where the pump is in continuous operation, the piston packing should be frequently

Figs. 1,247 to 1,251.—Metal valve with screw seat details. Fig. 1,247, screw seat; fig. 1,248 stud; fig. 1,249, metal valve; fig 1,250, spring; fig. 1,251, assembly.

adjusted, especially on high pressure systems. A small leak here will occasion considerable waste of fuel.

Leakage at the valves may be caused by improper seating due to uneven wear Valves should be frequently examined and refaced or turned over when worn.

The back flow loss may result from excessive speed, or improper adjustment of the valve springs. In general, the higher

the speed, the stronger should the spring be to insure rapid closing of the valves. The strength of spring should be governed somewhat by the lift.

For instance, when the lift is excessive, there will be very little difference in pressure above and below the valve, hence the available pressure for opening the valve will be small. In this case a very stiff spring might offer enough resistance to prevent the opening of the valve, or reduce the opening to such an extent that the water does not enter fast enough to fill the

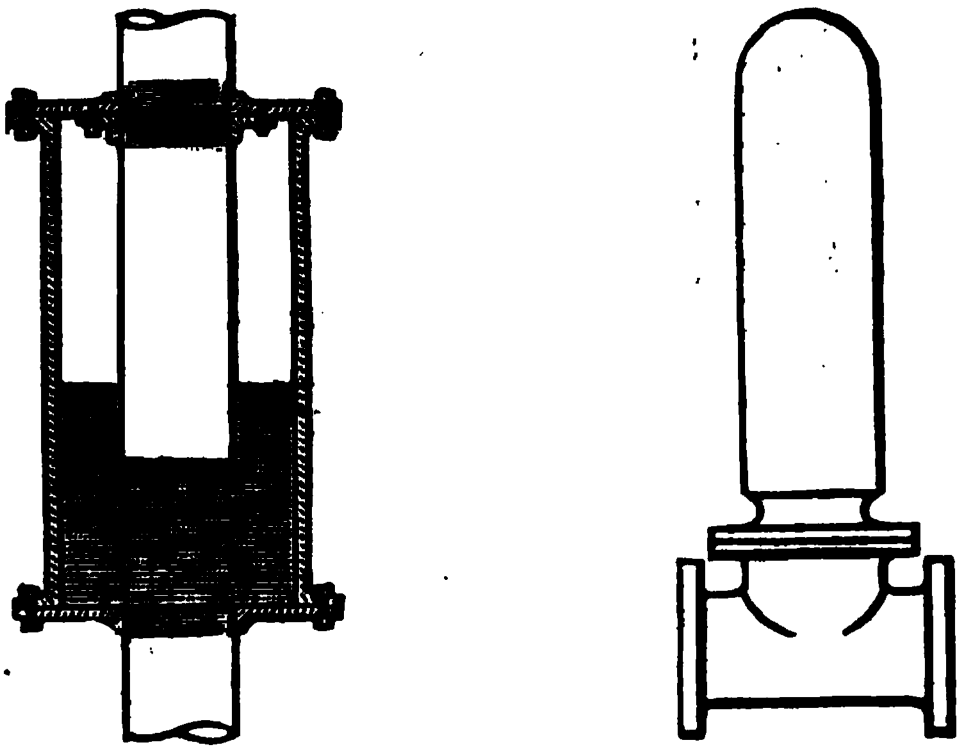
FIG. 1,252.—The air chamber. This part of the pump has been compared to the fly wheel of an air engine. Energy is here stored and imparted to the discharge column of water, thus keeping it in motion and compensating for the intermittent action of the pump.

cylinder at each stroke, consequently only a part of the cylinder displacement is discharged.

The Air Chamber.—The object of the air chamber is to make the flow of water more nearly continuous, and to reduce the shock of the impact at the beginning of the stroke, caused by the intermittent discharge.

Fig. 1,252 shows an air chamber placed on the delivery end of

a single acting plunger pump. When the inward stroke of the plunger takes place, the lower part of the water discharged from the cylinder goes to the delivery pipe, but a portion enters the air chamber and compresses the air in the upper part of the chamber. On the return stroke, when no water is passing out of the delivery valve the excess of pressure in the air chamber, due to the previous compression of the air therein, is sufficient to force the water out into the delivery pipe, and thus keep the water in this pipe in motion.



FIGS. 1,253 and 1,254.—Two types of vacuum chamber. The one shown in fig. 1,253 should be placed in such position as to receive the impact of the column of water in the inlet pipe, and be of considerable length rather than of large diameter and short.

There seems to be no standard for sizes of air chambers on pumps. A large air chamber permits running the pump at an increased speed.

A defect in the operation of air chambers is caused by the action of the water in absorbing air. Air chambers, therefore, are usually built with a small neck or entrance so that only a small surface of water is exposed to the air.

For heavy pressures particularly, air chambers should be arranged so that they can be recharged with air to replace that absorbed by the water.

Air or vacuum chambers are sometimes placed on the suction end, especially where there is considerable length of intake pipe. The action of a vacuum chamber is practically the reverse of that of the air chamber.

In operation the air pressure is less than the pressure in the intake pipe, thus tending to keep the column of water in motion during the intermittent action of the pump.

Vacuum chambers are of two types as shown in figs. 1,253 and 1,254. The one shown in fig. 1,253 should be placed in such position as to receive the impact of the column of the water in the inlet pipe and of considerable length rather than of large diameter and short. The size of the neck is substantially the same as in the air chamber.

Fig. 1,255 shows the water end of a pump fitted with air and vacuum chambers.

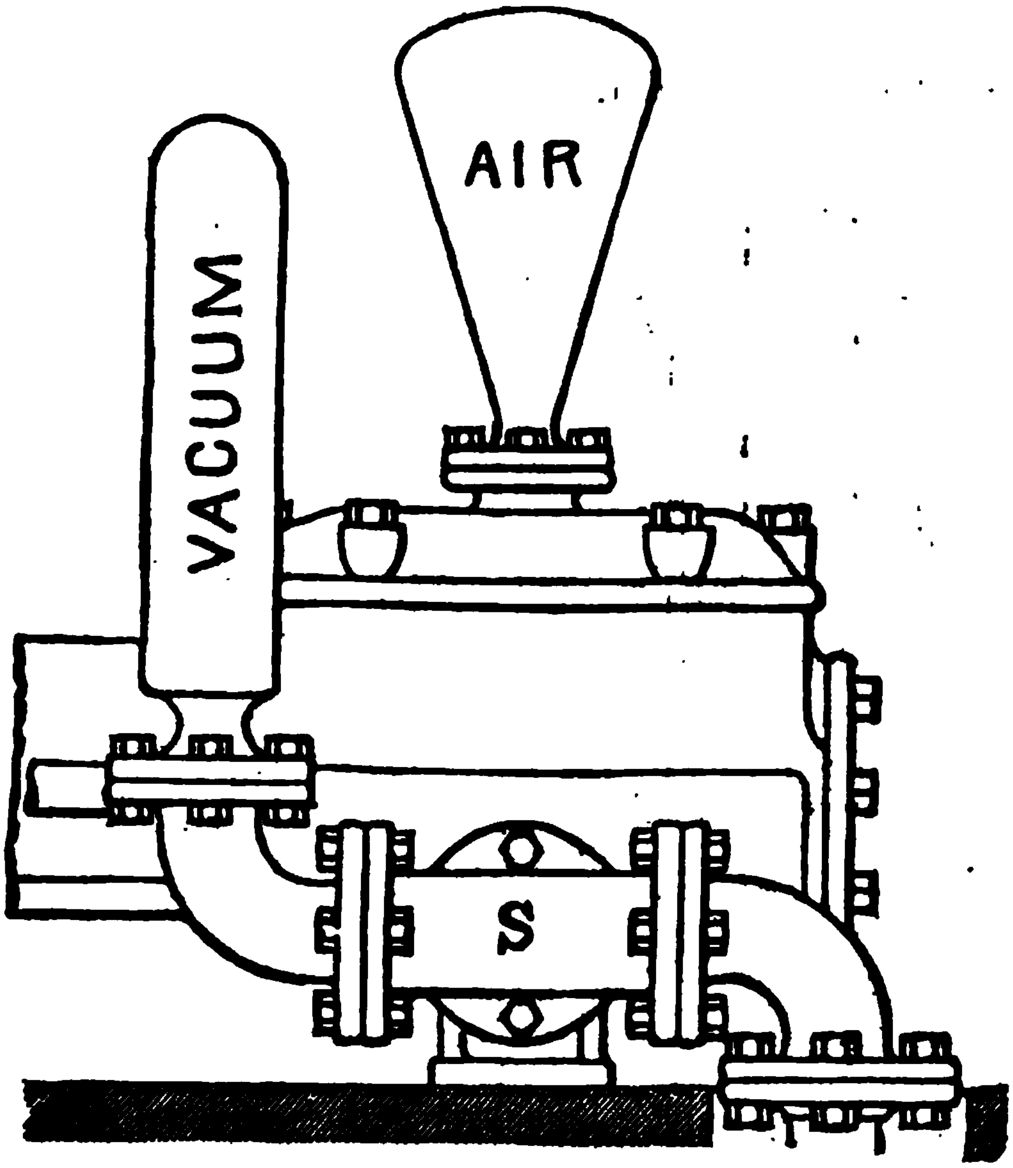


FIG. 1,255.—Water end of a horizontal pump, showing location of air and vacuum chambers.

The capacity of the air chamber varies in different makes of pump from 2 to 3½ times the volume of the water cylinder in single cylinder pumps, and from 1 to 2½ times the volume of the water cylinder in the duplex type. *The volume of the water cylinder is represented by the area of the water piston multiplied by the length of stroke.*

NOTE.—In large pumping plants small air pumps are employed for keeping the air chambers properly charged. In smaller plants an ordinary bicycle pump and a piece of rubber tubing are used to good advantage.

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Direct Acting Pumps with Equalizers.—Various devices have been fitted to direct acting pumps, in order to enable them to use steam expansively, and thus work with greater economy. If in the ordinary pump, steam be cut off before the end of the

Fig. 1,257.—Steam end of a Cornish mine pump. The considerable size of these pumps is shown by the dimensions, and emphasized by relative size of the man standing at the left of the fly wheel.

stroke, the pressure on the piston will gradually fall as expansion takes place. Hence, the power at the steam end may be so reduced before the completion of the stroke that it will fail to overcome the resistance at the water end, and the pump will therefore stop. In order to overcome this difficulty, some

device, such as shown in fig. 1,258, must be provided to equalize or compensate for the variable pressure at the steam end. A pump having this improvement is shown in fig. 1,259, it is of the tandem compound duplex type, as used to some extent in water works.

Horizontal Fly Wheel Pumps.—Instead of compensating cylinders for equalizing the variable pressure on the steam piston due to early cut off, a fly wheel is frequently used. There are numerous designs of this type of pumping engine differing

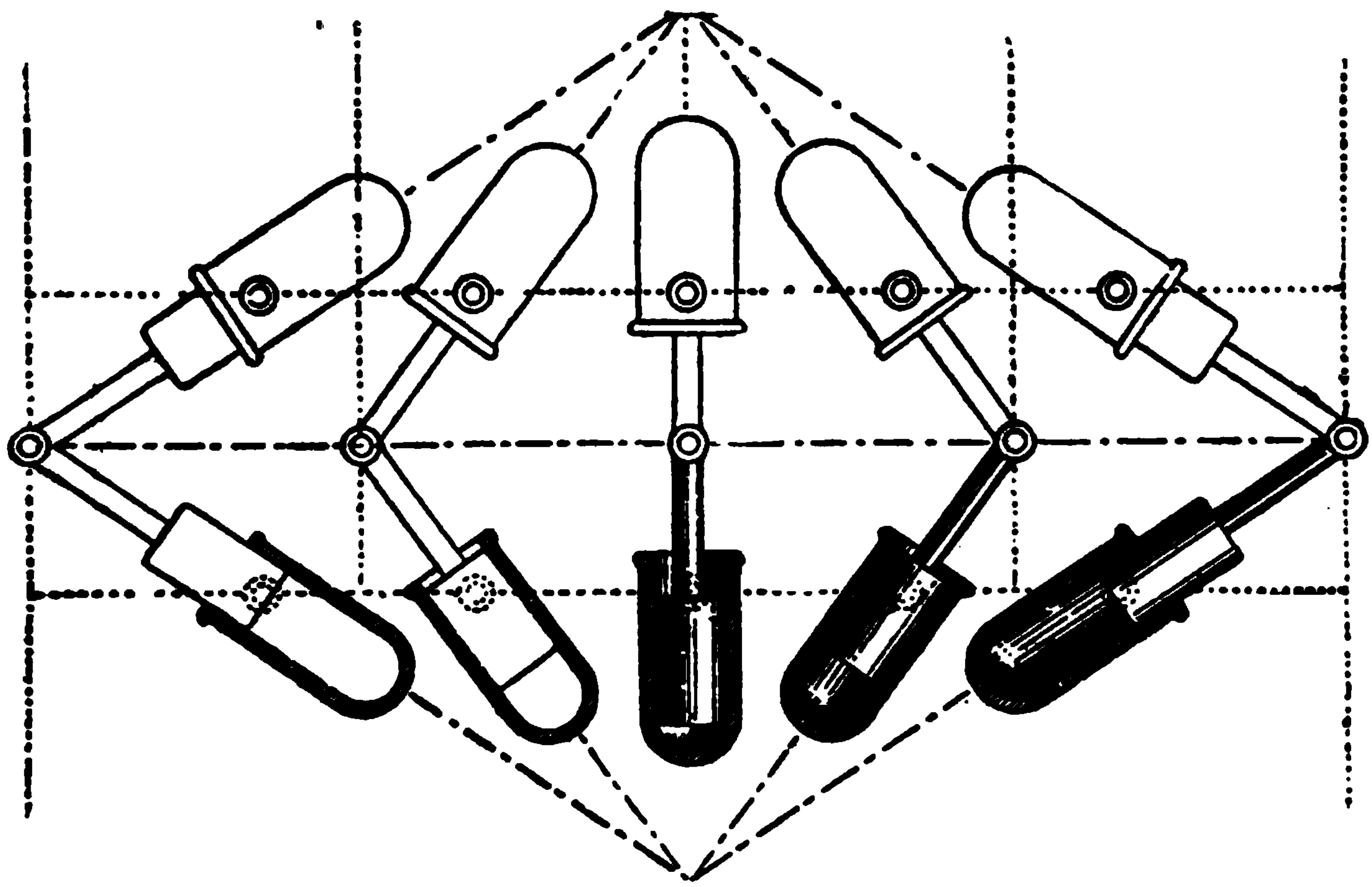

FIG. 1,258.—Compensating cylinders of the Worthington pump; view showing the cylinders in several positions during the stroke. The object of these cylinders is to equalize the pressures at the steam and water ends of the pump, thus permitting an early cut off in the high pressure cylinder, to secure greater economy in the use of steam.

greatly in the mechanical details and arrangement of parts. An example of the horizontal fly wheel pump is shown in fig. 1,260.

In this design the low pressure piston rods are directly in line with the pump plungers, the high pressure cylinder lying on top of the low pressure.

A swinging beam with links connects the cross head of both engines,

starts its stroke, these plungers are driven into their shell, compressing the air before them, the whole swinging on trunnions as the pump rod advances. When the pump has and steam is cut off by the valves, the auxiliary plunger assists to drive the rod on its way, thus giving under it at the commencement of the stroke.

FIG. 1,280.—The Holly horizontal compound pumping engine. An oscillating beam with links connects the high and low pressure pistons, and fly wheel. The strokes take place in unison from opposite ends of the cylinders.

so that the power developed by the high pressure engine which is not absorbed in driving the fly wheel is transmitted to the pump.

The high pressure steam is controlled by double beat valves worked by eccentrics off a cam shaft which lies at right angles to the crank shaft, being driven by bevel gearing.

Flat gridiron valves between the two cylinders serve both as exhaust valves for the high pressure and admission valves for the low pressure; separate grids being fitted for the final low pressure exhaust.

The air pump is driven off a vibrating arm on the beam gudgeon.

This engine is also built to operate as a non-compound engine, in which case the upper or high pressure cylinders and connections are omitted, and the lower steam cylinders are provided with automatic cut off valves. Steam is admitted to these cylinders direct from the boiler and exhaust into the condenser.

FIG. 1,261.—Three Morris direct connected dry dock centrifugal pumping engines. Capacity of each 38,333 gallons per minute, 350 horse power.

NOTE.—***Where conditions of head allow,*** centrifugal pumps directly connected to steam engines are used, as such outfits are self-contained and take up but little space. With very large pumps belt drive is prohibitive, and while occasionally a rope drive is resorted to, a directly connected outfit makes the most economical pumping unit. A steam driven Centrifugal pump is, under every condition, more economical in fuel than a reciprocating steam pump.

NOTE.—***The high speed of the steam turbine*** makes it possible to use direct connected centrifugal pumps for high head where steam is available. Pumps directly connected to reciprocating engines can only be used for comparatively low heads owing to the engine's limited rotative speed. On account of small space occupied, light weight and minimum of attention required, the steam turbine driven pump is very attractive for many situations.

NOTE.—***Installing centrifugal pumps.*** The suction elevation or lift should be made as low as possible, and also the suction pipe as short and direct as possible with a minimum number of elbows or bends. The suction pipe should be so arranged that no air pockets can form. Particular care should be taken that the suction pipe be air tight, as otherwise there will be difficulty with the pump's operation. The dynamic lift, that is, the actual elevation plus frictional resistances through suction piping, should, if possible, not exceed 25 feet, and preferably should be much less than this, especially in extremely high speed pumps. The larger the discharge pipe, and the fewer the number of elbows, the less the friction head and power required to drive the pump. The foundations need not be as heavy as for a reciprocating pump, but should be amply rigid and strong to properly support the pump and maintain correct alignment. Great care should be taken that the pump be properly leveled on the foundation, and that it be not finally bolted down until it has been carefully ascertained that shaft is properly in line and turns freely in the bearings. If the pump be directly connected to the driver through a coupling, the alignment of the coupling should be proven. The bearings should be thoroughly cleaned before the pump is started, and lubrication supplied. The shaft stuffing box or stuffing boxes should be properly packed with best quality of soft packing well greased and graphited, and gland should not be tightened up tight as that will burn the packing and cut the shaft. A slight water leakage from the stuffing boxes does no harm and is an indication that the packing is not too tight. With a suction lift, it is sometimes advisable to provide the stuffing boxes with water seals to prevent air leakage into the pump. Usually the water seal is supplied by water from the pressure portion of the pump shell, which is all right if the pump be handling clean water; otherwise water should be supplied from any other convenient source with pressure of at least 20 pounds.

Triple Expansion Pumping Engines.—For high economy in the use of steam, its expansion is frequently divided into three stages, a further division is ineffective in reducing the steam consumption unless exceptionally high initial pressure be used. A type of triple expansion pumping engine used to some extent, is shown in fig. 1,261.

Capacity of Pump.—This term relates to the amount of water a pump is able to deliver when operated at a specified speed. There are two kinds of capacity:

1. Theoretical capacity;
2. Actual or net capacity.

The theoretical capacity represents the pumping ability of a perfect pump, and is expressed as *the volume in cubic feet or gallons* ***displaced*** *by the pump per minute.*

Since it is impossible to construct a perfect pump, it is customary in computing capacity, to first calculate the theoretical capacity and then make allowance for the various losses due to slip, leakage, etc.

Ques. What kind of a pump will pump more than its theoretical capacity, and why?

Ans. A single acting lift pump having bucket valves, because the column of water does not cease flowing when the bucket descends, that is, especially at high speeds the foot and head valves remain open all the time, and the bucket valve accordingly under such conditions is the only valve essential to operation.

How to Figure Capacity.—*RULE: Multiply the area of the piston in sq. ins. by the length of the stroke in ins., and by the number of delivery strokes per minute, divide the product by* **1,728** *to obtain the theoretical capacity in* **cu. ft.,** *or by* **231** *to obtain theoretical capacity in* **U. S. gallons.** *The result thus obtained*

is to be multiplied by an assumed **factor** *representing the efficiency of the pump to obtain the* **approximate** *net capacity.*

The rule expressed as a formula is

$$\text{Approximate net capacity} = \frac{.7854 \times D^2 \times L \times N}{1{,}728} \times (1 - f) \textit{ cu. ft.,}$$

or

$$= \frac{.7854 \times D^2 \times L \times N}{231} \times (1 - f) \textbf{ gallons}$$

in which

D^2 = square of piston or plunger diameter in sq. ins.;
L = length of stroke in ins.;
N = number of delivery strokes per minute;
f = factor representing assumed slip in per cent. of displacement;
1,728 = cu. ins. in one cu. ft.;
231 = cu. ins. in one U. S. gallon.

Example.—What is the approximate net capacity of a 3×5 double acting power pump running at 75 revolutions per minute with an assumed slip of 5 per cent., applying this formula?

$$\text{Approximate net capacity} = \frac{.7854 \times 3^2 \times 5 \times 150}{1{,}728} \times (1 - .05) = 2.92 \text{ cu. ft.}$$

$$= \frac{.7854 \times 3^2 \times 5 \times 150}{231} \times (1 - .05) = 21.8 \text{ galls.}$$

Theoretical Horse Power at the Water End.—The theoretical horse power required to raise water at a given rate to a given elevation is obtained by the following formula:

$$\text{T. H. P.} = \frac{V \times W \times (L + H)}{33{,}000}$$

in which

V = volume in cu. ft. per minute;
W = weight of one cu. ft. of water;
L = lift in ft.;
H = head in ft.

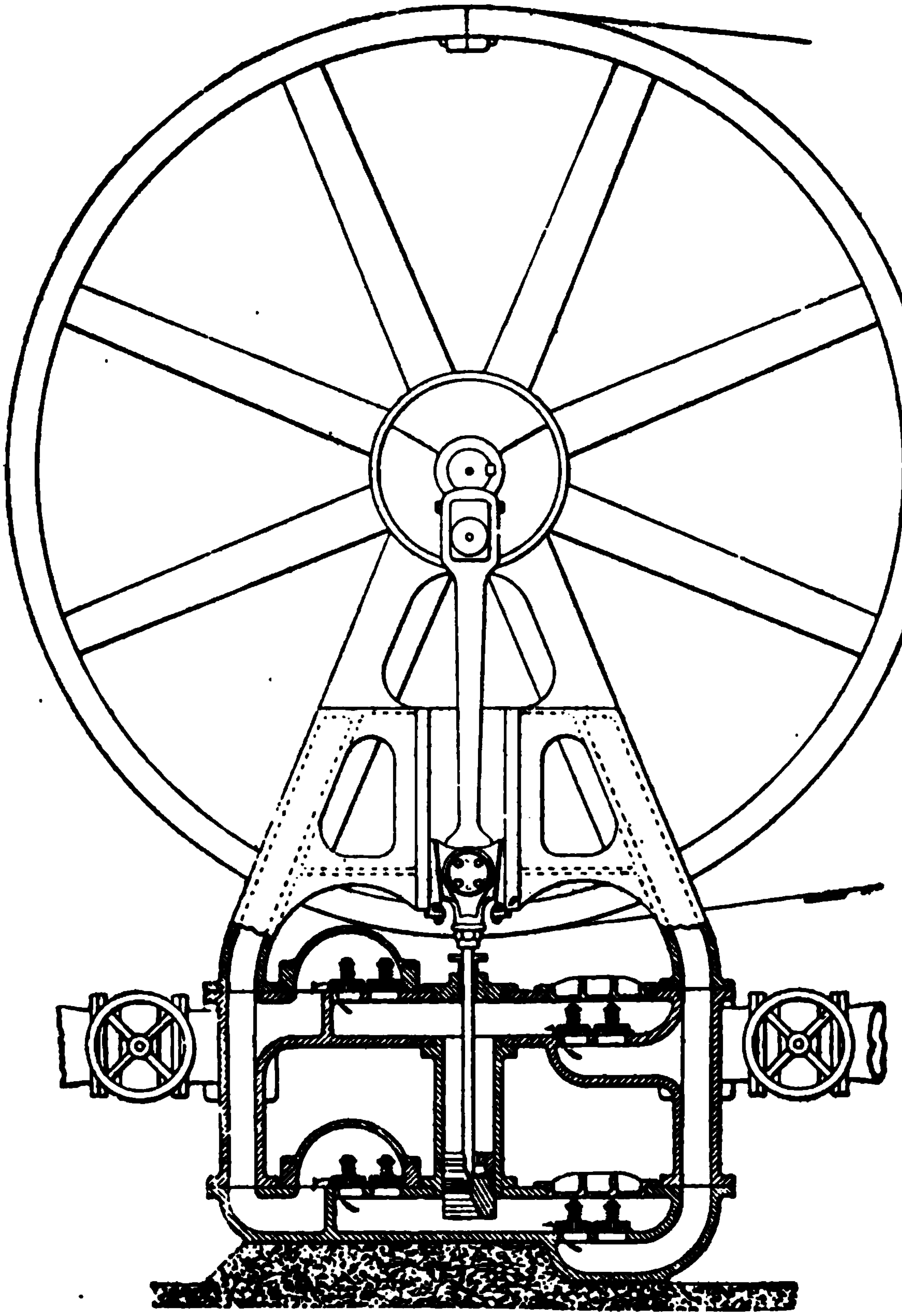

FIG. 1,262.—Dunham duplex double acting power pump. ***The general design*** comprises two pairs of horizontal water chambers, each pair connected at each end by vertical water passages, and at the center by a vertical working barrel or cylinder. The upper water chambers of each pair are connected at each end by arms firmly bolted together in a vertical plane. passing lengthwise through the center of the pump. In each arm there is an intake or discharge opening where a gate valve can be placed. By closing one pair of gate valve (one intake and one discharge) the water can not flow from one pair of chambers to the other, at either end, thus making it convenient to use only one side, or one-half the pump in case of repairs. Each water chamber carries two sets of valves, intake and discharge; one on each side of the vertical cylinder. The valve decks proper are cast as part of the water chamber, but each deck carries one bronze plate in valve rest. The valve chamber openings are provided with large covers making valves readily accessible. Each upper water chamber supports an A frame of box section forming air chambers for both intake and discharge. The two A frames carry the main bearings for the shaft. The drive is by belt instead of gearing, thus eliminating noise and friction of extra bearings.

Example.—What is the theoretical horse power required to raise 100 cu. ft. of water 200 ft., with a 10 ft. lift when the water is at a temperature of 75° Fahr., and when at 35° Fahr.?

For a temperature of 75°, one cu. ft. of water weighs 62.28 lbs. Substituting this and the other data in the formula,

$$\text{T.H.P.} = \frac{100 \times 62.28 \times (10+200)}{33{,}000} = 39.63$$

Now if the water have a temperature of only 35°, as might be in very cold weather, the weight of one cu. ft. will increase to 62.42, and the horse power would accordingly increase in proportion to the ratio of the two weights, or

$$\text{T.H.P. (at 35° Fahr.)} = 39.63 \times \frac{62.42}{62.28} = 39.7$$

By observing the very slight difference in the two results it will be seen that for ordinary calculation, the temperature need not be considered, taking the usual value 62.4 lbs.

Horse Power Absorbed at the Water End.—The actual horse power required at the water end of a pump (not including slip or mechanical efficiency) is equal to *the theoretical horse power plus an allowance for the friction of the water through the pipes and pump passages.*

There is also friction of water in the elbows which is usually taken into account. Values for these two items are obtained by consulting tables of friction of water in pipes from which the virtual head to be used is easily found and which when inserted in the T.H.P. formula will give the "actual horse power" as above defined.

Duty of Pumps.—The word "duty" is used in engineering to express the efficiency of a steam pumping engine as measured by the work done by a certain quantity of fuel, or steam. Duty, then, stands for foot pounds or work done, and means the number of pounds of water lifted one foot, or its equivalent, by 100 pounds of coal, or 1,000 pounds of saturated steam.

Table of Theoretical Horse Power Required to Raise Water to Different Heights

Gallons per Minute	5 feet	10 feet	15 feet	20 feet	25 feet	30 feet	35 feet	40 feet	45 feet	50 feet	60 feet	75 feet	90 feet	100 feet	125 feet	150 feet	175 feet	200 feet	250 feet	300 feet	350 feet	400 feet	Gallons per Minute
5	.006	.012	.019	.025	.031	.037	.044	.05	.06	.06	.07	.09	.11	.12	.16	.19	.22	.25	.31	.37	.44	.50	5
10	.012	.025	.037	.050	.062	.075	.087	.10	.11	.12	.15	.10	.22	.25	.31	.37	.44	.50	.62	.75	.87	1.00	10
15	.019	.037	056	.075	.094	.112	.131	.15	.17	19	.22	.28	.34	.37	.47	.56	.66	.75	.94	1.12	1.31	1.50	15
20	.025	.050	.075	100	.125	.150	175	.20	.22	.25	.30	.37	.45	50	.62	.75	.87	1.00	1.25	1.50	1.75	2.00	20
25	.031	.062	.093	.125	.156	.187	.219	.25	.28	.31	.37	.47	.56	.62	78	.94	1.09	1.25	1.56	1.87	2.19	2.50	25
30	.037	.075	.112	.150	.187	.225	.262	.30	.34	.37	.45	.56	67	.75	.94	1.12	1.31	1.50	1.87	2.25	2.62	3.00	30
35	.043	.087	.131	.175	.219	.262	.306	.35	.39	.44	.52	.66	.79	.87	1.08	1.31	1.53	1.75	2.19	2.62	3.06	3.50	35
40	.050	.100	.150	.200	.250	.300	.350	.40	.45	.50	.60	.75	.90	1.00	1.25	1.50	1.75	2.00	2.50	3.09	3.50	4.00	40
45	.056	.112	.168	.225	.281	.337	.394	.45	.51	.56	.67	.84	1.01	1.12	1.41	1.69	1.97	2.25	2.81	3.37	3.94	4.50	45
50	.062	.125	.187	.250	.312	.375	.437	.50	.56	.62	.75	.94	1.12	1.25	1.56	1.87	2.19	2.50	3.12	3.75	4.37	5.00	50
60	.075	.150	.225	.300	.375	.450	.525	.60	.67	.75	.90	1.12	1.35	1.50	1.87	2.25	2.62	3.00	3.75	4.50	5.25	6.00	60
75	.093	187	.281	.375	.469	.562	.656	.75	.84	.94	1.12	1.40	1.69	1.87	2.34	2.81	3.28	3.75	4.69	5.62	6.56	7.50	75
90	.112	.225	.337	.450	.562	.675	.787	.90	1.01	1.12	1.35	1.68	2.00	2.25	2.81	3.37	3.94	4.50	5.62	6.75	7.87	9.00	90
100	.125	.250	.375	.500	.625	.750	.875	1.00	1.12	1.25	1.50	1.87	2.25	2.50	3.12	3.75	4.37	5.00	6.26	7.50	8.75	10.00	100
125	.156	.312	.469	.625	.781	.937	1.094	1.25	1.41	1.56	1.87	2.34	2.81	3.12	3.91	4.69	5.47	6.25	7.81	9.37	10.94	12.50	125
150	.187	.375	.562	.750	.937	1.125	1.312	1.50	1.69	1.87	2.25	2.81	3.37	3.75	4.69	5.62	6.56	7.50	9.37	11.25	13.12	15.00	150
175	.219	.437	.656	.875	1.093	1.312	1.531	1.75	1.97	2.19	2.62	3.28	3.94	4.37	5.47	6.56	7.66	8.75	10.94	13.12	15.31	17.50	175
200	.250	.500	.750	1.000	1.250	1.500	1.750	2.00	2.25	2.50	3.00	3.75	4.50	5.00	6.25	7.50	8.75	10.00	12.50	15.00	17.50	20.00	200
250	.312	.625	.937	1.250	1.562	1.875	2.187	2.50	2.81	3.12	3.75	4.69	5.62	6.25	7.81	9.37	10.94	12.50	15.72	18.75	21.87	25.00	250
300	.375	.750	1.125	1.500	1.875	2.250	2.625	3.00	3.37	3.75	4.50	5.62	6.75	7.50	9.37	11.25	13.12	15.00	18.75	22.50	26.25	30.00	300
350	.437	.875	1.312	1.750	2.187	2.625	3.062	3.50	3.94	4.37	5.25	6.56	7.87	8.75	10.94	13.12	15.31	17.50	21.87	26.25	30.62	35.00	350
400	.500	1.000	1.500	2.000	2.500	3.000	3.500	4.00	4.50	5.00	6.00	7.50	9.00	10.00	12.50	15.00	17.50	20.00	25.00	30.00	35.00	40.00	400
500	.625	1.250	1.875	2.500	3.125	3.750	4.375	5.00	5.62	6.25	7.50	9.37	11.25	12.50	15.62	18.75	21.87	25.00	31.25	37.50	43.75	50.00	500

This table gives the actual water horse power. When selecting motors, turbines allowance must be made for pipe friction and loss in the pump, gears, belts, etc. One foot head equals .43 pounds pressure to the square inch.

Formerly duty was expressed on the coal basis, but this has fallen into disuse owing to the variations in the quality of the latter.

Duty expressed per 1,000 lbs. of steam is equivalent to 100 lbs. of coal when the evaporation is 10 to 1. This result is readily obtainable with good grades of coal when the boilers are correctly proportioned and in proper working condition.

Duty per 100 lbs. of coal would show the combined efficiency of the pump and boiler; when expressed on the steam basis, the efficiency of the pump alone is obtained. The latter, therefore, is generally used, as the result sought is to determine how economical the pump is in the use of steam.

Another unit of duty is, the *foot pounds of work at the water end per million heat units furnished by the boiler.* This is the equivalent of 100 pounds of coal where each pound imparts 10,000 heat units to the water in the boiler, or where the evaporation is $10{,}000 \div 965.7 = 10.355$ pounds of water from and at 212°, per pound of coal.

The last mentioned unit which was reported in 1891 by a committee of the *A. S. M. E.* (*Trans.* XII, 530), reaffirmed it in 1915 as the standard unit and defined it as follows: ***the duty per million heat units is found by dividing the number of foot pounds of work done during the trial by the total number of heat units consumed, and multiplying the quotient by*** 1,000,000. The amount of work is found in the case of reciprocating pumps by multiplying the net area of the plunger in sq. ins., the total head in lbs. per sq. in.* by the length of the stroke in feet, and the total number of single strokes during the trial; finally allowing for the percentage of leakage of the pump. In cases when the water delivered is determined by weir or other measurement, the work done is found by multiplying the weight of water discharged during the trial by the total head in feet.

*NOTE.—The total head is determined by adding together the pressure shown by the gauge on the force main, the vacuum shown by the gauge on the suction main, and the vertical distance between the center of the force main gauge and the point where the suction gauge pipe connects to the suction main, all expressed in the same units (pounds per sq. inch or sq. foot). A pet cock should be attached to the gauge pipe below each gauge cock, and opened occasionally so as to free the pipe of air in the case of the force main gauge, and of water in the case of the suction gauge. If the suction main be under a pressure instead of a vacuum the suction gauge should be attached at such a level that the connecting pipe may be filled with water when the pet cock is opened, in which case the correction for difference in elevation of gauges is the vertical distance between the centers of the gauges, and the reading of the suction gauge is to be subtracted from that of the force main gauge. If the water be drawn from an open well beneath the pump, the total head is that shown by the force main gauge corrected for the elevation of the center of the gauge above the level of water in the pump well. If there be a material difference in velocity of the water at the points where the two gauges are attached, a correction should be made for the corresponding difference in *velocity head.*

CHAPTER 21
SPECIAL RECIPROCATING ENGINES

Single Acting Engines.—For some purposes engines of this type can be used to advantage. Since the pressure of the steam acts only in one direc-
e bearings do
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n jacketted oscillat-
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dmitted through M,
n annular ring lower
ston through entire
d stroke, steam is
d through M and S,
side of piston and
ded through entire
; *3rd stroke,* steam
sted through S, to
st pipe. *Jacket*
enters at L, and
through upper head
cylinder jacket, the
densate being dis-
rged through F. The
re shows author's
6 in. stroke engine
proportioned for 8
using saturated
derate pressure, and
r high speed light
duty service, operating with about the same economy as a marine compound engine.

adjusted as close nor as often as with double acting engines, hence, they may be operated with less attention.

The characteristics of single acting engines are: minimum liability to get out of order or to pound, as the working parts are

FIGS. 1,264 to 1,266.—Acme two cylinder single acting engines for small unit high speed service. The sizes range from 2½×3½ to 7×7 developing at 80 lbs. initial pressure, 1¼ to 27 horse power, 50 lbs. M.E.P., 400 R.P.M. Any size may be operated up to 600 R.P.M. The valve is of the rocking type. It is a one-piece casting ground to fit a bored chamber which serves as the valve seat and cylinder head. The lubrication is by the splash system. The crank case is filled with water to such a height that the cranks just dip into it; a small quantity of heavy lubricating oil is then added, so that at every dip of the cranks the mixture is splashed to every part of the interior, where it collects in small pockets and flows in a continuous stream over all bearings.

reduced to the smallest number consistent with economy in the development of power; smooth and quiet operation at high speeds.

FIG. 1,987.—Westinghouse two cylinder

pin being thrown across the shaft through the action of the governor, thus varying the travel of the valve. The steam enters at U, at the center of the valve H, and exhausts at the ends through the exhaust pipe R. The main bearings E, are lubricated by the oil cups Q, from the outside; all drips from the main bearings run through the channels *r*, into the oil chamber B. Although the manufacture of this engine has been discontinued it is here shown, to illustrate **steam distribution** *in two single acting cylinders, by a single piston valve*, and because of the large number of these engines still in use.

They are usually made with two cylinders, cranks at 180 degrees, thus requiring only one valve which reduces the amount of valve gear parts required. They are suitable for small powers

gine with valve self-contained on the cylinder. In construction a rocker arm, A, one end of which is connected to the valve stem, is pivoted a block running in a quadrant, N, which is operated by

rom the difficulty of getting an efficient apparatus for an early cut off of steam, and the liability of the trunnions to leak; the former is the real difficulty, the latter existing more in imagination than practice. The solution has been found in the introduction of compound cylinders. The most successful oscillating engines, however, have been those of large power, working with a steam pressure not exceeding 30 lbs. per sq. in., and consuming on the average 2¾ lbs. of coal per indicated horse power per hour."

In construction, the cylinder is swung on trunnions, the piston rod being connected directly to the crank shaft, avoiding guides, cross head and connecting rod. There are two general classes of oscillating engines: 1, those with valve on the cylinder, as in fig. 1,268, and 2, those with ports oscillating about a stationary seat as in fig. 1,269. Either may be single or double acting.

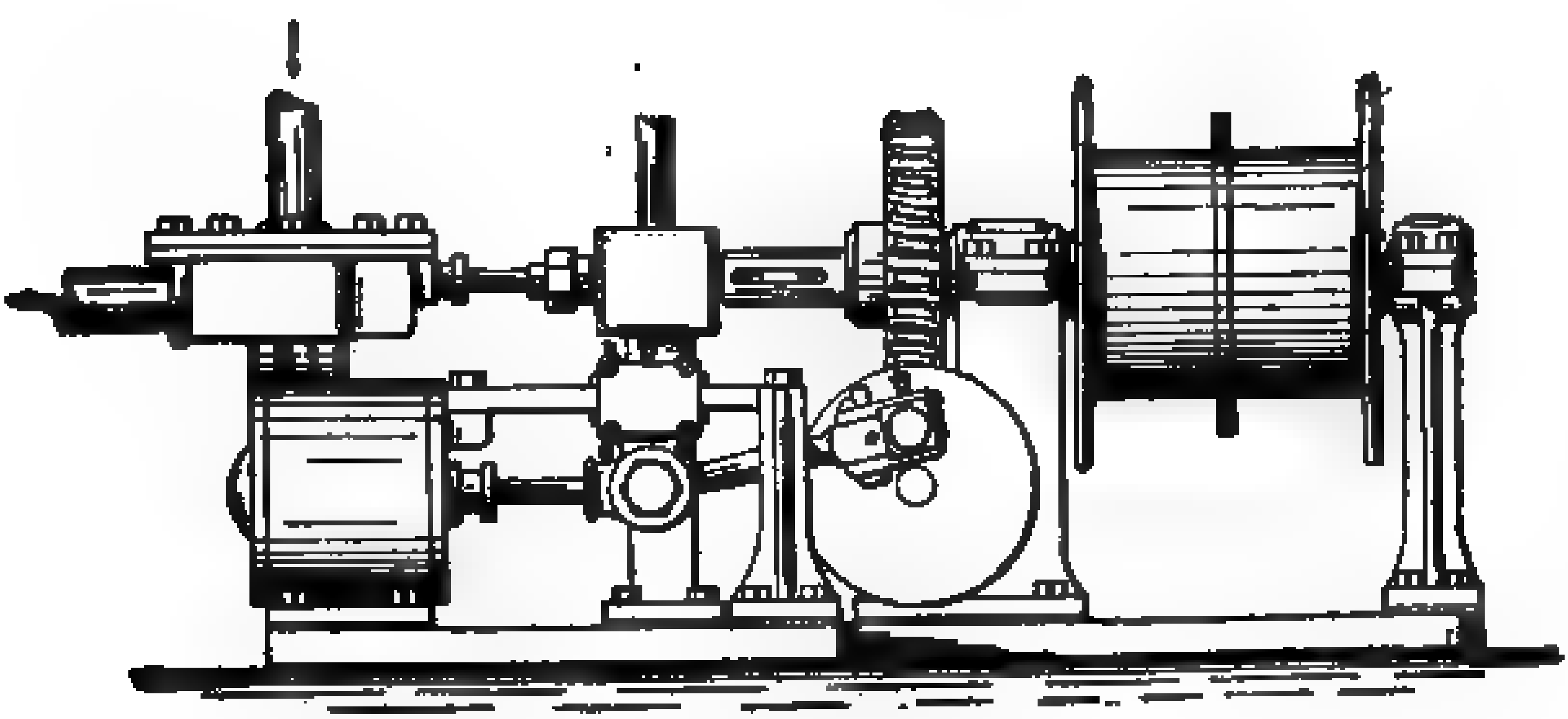
1,269.—Kriebel oscillating marine engine. ***In con- ruction,*** the engine frame OO, is made in one piece ad has boxes on each side to receive the crank shaft M, ad the solid trunnions E, which project at right igles from the upper head of the cylinder and on hich the cylinder is supported and pivoted. The ston H, is connected with the piston rod I, to the ank pin L, and the three are always in a straight line, nsequently as the piston moves up and down, the linder vibrates back and forth on the trunnions. The lve D, is a hollow, cylindrical casting enclosed in a sting A, attached to the engine frame. The bottom the valve has a smooth concave surface, while the oper end of the cylinder F, has a smooth convex surface. he two surfaces make a steam tight joint, and any ear that occurs is automatically taken up by springs , coiled around bosses above the valve. The steam id exhaust pipes S and T, connect with two brass bes R and R′, which are screwed into the valve and mmunicate with the valve ports X and QQ′. There e two cylinder ports P and P′, which open into the top id bottom of the cylinder. ***In operation,*** as the linder vibrates back and forth on the trunnions, the linder ports alternately take steam from the central lve port X, and exhaust through the ports Q and Q′. ***In reversing,*** the direction of the steam in the tubes can be changed so the cylinder ports will either take steam from the port X, and exhaust through the port Q and Q′, as above, or else take steam from Q and Q′, and exhaust through X, and thus reverse the engine. The piston rod has a long stuffing box N. The upper ends of the tubes R and R′, are received by fixed stuffing boxes, B, represents a counter-balance, which is bolted to the cranks of engines with 5×6 in. cylinder and upward. The reversing valve on top answers also as a throttle, as by moving the lever to a central position, the steam and exhaust ports are closed and the engine stops.

FIG. 1,268.—*Text continued.*

the eccentrics, E,E. The cylinder C, and piston rod vibrate the same as the connecting rod on other engines, the cylinder being suspended on both sides by trunnions T, which also serve to convey the steam to the valve chest, and the exhaust to the condenser. The pipes are inserted into the openings O, the joint being made steam tight by means of packing rings, allowing the cylinders to oscillate. The steam passes through a channel in the cylinder casting to the valve chest V, and is controlled by a slide valve in the ordinary way. The reversing gear may be of the loose eccentric type, or the *so called* Stephenson link motion, as shown. The eccentric rods are connected to the link L, which slides over the link block attached to a pin in the quadrant N, and by means of the reach rod R, and the reversing gear shown in the figure, either eccentric may be made operative.

In the second type the trunnions are used as steam passages. The trunnions have interposed between them and the cylinder body a belt, which conveys the steam to and from the valve boxes. An engine of this type with reversing gear is shown in fig. 1,268.

Steering Engines.—The steam steering engine serves the purpose of turning the rudder of the ship more powerfully and



FIGS. 1,272 and 1,273.—Plain and side view of a steering engine showing construction.

rapidly than can be done by hand. Its power is applied through rope, chain or gear transmission to a lever or sheave on the rudder stock. But its operation is intermittent and is also

peculiar, in that it swings the rudder only through a part of the arc, less than 90 degrees, while the engine makes several revolutions

FIG. 1,274.—Steam steering engine shown connected to the pilot house wheel. The pilot wheel is connected with the regulating valve of the engine by means of a sprocket chain and screw. *In operation*, when the pilot wheel is revolved, the screw, also being revolved, will draw or push the regulating valve, to open a port, which will admit steam to run the engine in the direction necessary to turn the rudder the way desired, but the nut in which the screw works, is attached to the drum shaft, which transmits the motion of the engine by cable or chain to the rudder, and when the drum revolves, thus also turning the nut, the screw, which is attached to the stem of the regulating valve is forced in the opposite direction, thus shutting off the steam. It will be understood, that the engine only revolves while the wheel is turned, thus the rudder can be turned any desired amount. The action of the steering engine, with its mechanism is somewhat similar to the steam reversing gear. The gear connection between the pilot wheel and the drum shaft, here shown, is for the purpose of operating the rudder directly by the pilot wheel if desired.

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ports through the
bridge and cl 1 of
the rudder is

FIG. 1,275.—Self-centering mechanism of the Williamson steering engine. In a steering engine it is necessary to have the steam valve controlled automatically so that the movement produced by the engine is kept proportional to the movement given the steering wheel. The engine controlling valve starts to move at once in response to any movement of the steering wheel in either direction. The engine keeps moving only so long as the control valve is kept in motion by the steering wheel. To function in this manner motion is given to the shaft A, from the rotation of the steering wheel in the pilot house. The threaded shaft B, which is prevented rotating by the follow-up gears E and F, when the engine is at rest, is pulled endwise by the rotation of miter gears C and D, upon rotation of shaft A. Gear D, which is threaded to fit shaft B, is held longitudinally by the bearing G. This makes it a block through which the shaft B, is drawn. The endwise motion of shaft B, controls the opening of the change valve which admits steam to the main valves of the engine cylinders. Steam having been admitted to the cylinders during the turning of the shaft A, the revolving of the drum shaft rotating the follow-up gears E and F, returns the change valve to a neutral position, when the shaft A, ceases to turn, thereby stopping the steam flow to the main valve and causing the steering engine to come to a stop. It will thus be seen that the movement of the steering wheel in the pilot house admits steam to the engine, while the motion of the engine itself cuts off the steam. Stops are fitted to the automatic attachment to regulate the number of turns of the wheel from "hard over" to "hard over" These stops are adjustable to agree with the distance the rudder should travel to prevent bringing up against the chocks at the "hard over" position. Thus undue strains on the chains and connections between the rudder quadrant and the engine are prevented. The steam steering wheel, made of small diameter, is used only for opening the engine valve. The number of turns of the wheel, from *hard a port* to *hard a starboard*, varies from six to ten, according to the size of the vessel. The steering wheel is mounted on a column or pedestal of brass or iron. The wheel is connected with the valve motion of the engine by means of shafting and gearing. A tell-tale or indicator is located on the top of the column, showing the angle through which the quadrant or rudder has moved.

The steering gear consists of two complete engines, with regular valves, which, however, have no angular advance, lead or laps. The two cranks are set at 90°, to insure a positive starting in any position. The construction of the engines should be strong and durable, with large bearing surfaces, as the strains upon them may be

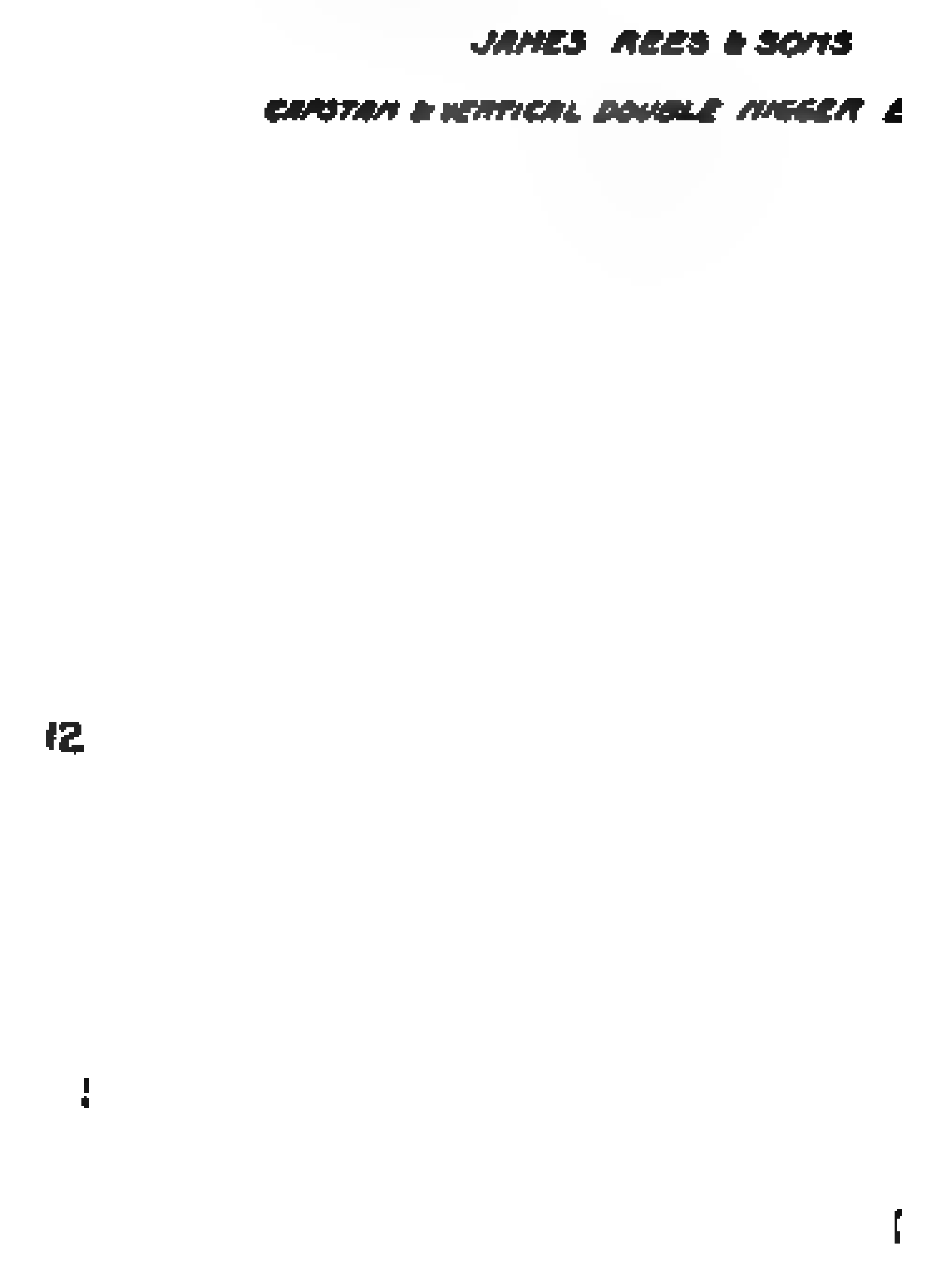
FIG. 1,276.—Worm of Williamson worm drive steering engine. It is a double worm with adjustable bearings. The teeth of both worm and wheel are so shaped that the entire face of the worm is in contact with the wheel at the same time thereby securing the maximum bearing surface and the minimum of wear. The worm, W W, is of steel or brass, according to size of engine, and is driven from the crank shaft by feathers, allowing a sliding motion. The adjustable bearings, BB, take the thrust, independent of the main journals of the crank shaft, and take up the lost motion due to any war of the worm or wheel. This prevents any back-lash or noise in reversing the engine. Adjustment is accomplished by slacking the bolts D D, and tightening the screws S S. To determine whether the adjustment is too tight, turn over the engine by hand, by means of the crank wheel. Adjustment should be made as soon as any lost motion is discovered, so that the teeth may always be kept in proper bearing. Excessive lost motion should be taken up by gradual adjustment, not all at once. The worn worm is self-lubricated by means of the oiling attachment, L. It consists of a wheel revolving in an oil reservoir, and always in contact with the worm.

quite severe in a sea way. The steam steerer is placed directly under the pilot house, in the engine room or near the rudder stock. The transmitting gear needs attention in keeping it taut, and in providing that the position of the hand wheel and its indicator corresponds to the position of the rudder.

Steam Reversing Gear.—On large engines the valve gear

parts which have to be moved to reverse position to change the direction of motion of the engine have considerable weight and cannot be moved by hand without considerable exertion, thus steam reversing gear is used. The form of reverse gear most commonly employed in America is of the so called "floating" or "self centering" types as shown in fig. 1,277, its operation being explained in the text under the figure.

g gear, used with
with the Stephen-
column D, is at-
C, which actuates
n of this bell-crank
link R, which is
connected, as shown, to the shifting link L. The valve of the cylinder C, is controlled by the hand lever, H; and when this is turned so as to admit steam to the bottom face of the piston, the bell crank B, and link R, describe the paths indicated by the dotted arcs, thus shifting link L, to the left, and reversing the action of the eccentric rods EE, on the valve stem S, of the main cylinder. ***Other lettered parts are;*** T, the tumbling shaft on which B, is pivoted; G, guide rod on which the cross head F, slides; I and I, cushion springs against which F, strikes, thus preventing the piston of C, striking the cylinder heads.



FIGS. 1,278 to 1,280.—Rees capstan and vertical double "*nigger*" engine. The counter-shaft and double purchase with the double 6×8 piston valve reversing en to the capstan by vertical and horizontal shafts with bevel gears and pinions as shown. *R.p.m.* of engine to one of capstan: with counter-shaft thrown in, 213; thrown out, 18.

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and the crank pi
closed. Steam

FIGS. 1,881 and 1
casting, which
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which register

It is evident from the figure and explanation that the mechanism will operate the reverse lever arm synchronously with the control lever but in opposite direction. Should the weight of the moving parts or expansion of the steam carry the piston beyond the descend point, the valve would also be displaced from its neutral position admitting steam on the other side and thus establishing equilibrium.

Ques. What peculiar action is present in engines equipped with steam reversing gear?

Ans. Since the elastic force of steam is used to "lock" the gear in the running position, when the mechanism is vertical as shown, the parts tend to move downward until the valve begins to admit steam which then drives the parts back to the desired position. This vibration causes a change in the point of cut off which causes the engine to periodically slow down and speed up.

Ques. Why is lap given to the valve.

Ans. To reduce the sensitiveness of the gear.

Very little lap is given, about 1/16 steam and 1/8 exhaust, the latter being added to cushion the parts.

Square Piston Engine.—This type of engine is shown in the accompanying cuts. Reduced to its simplest elements it consists of simply two movable pieces, one sliding inside the other,

NOTE.—***Turning gear.*** The object of such gear is to turn the engine by external power, as when overhauling, setting the valves, such gear being used for engines, too large to be turned by hand. The turning gear usually consists of a large worm wheel placed on the main shaft just aft of the bed plate, geared through worm and spur gearing to a small engine, usually a double simple engine with cranks at 90°. ***The gearing ratio*** is such that many hundred revolutions of the turning engine may be required to one of the turning wheel or main engine shaft. This gear must be so arranged as to be readily thrown in and out of connection with the main turning wheel. This is usually accomplished by carrying the main worm on a shaft which is pivoted, and which can thus be locked in either of two positions, in one of which the worm is in gear, and in the other, out of gear, or else by driving the worm on a shaft with a feather, thus providing for endwise motion, and for fixing it in either of two locations on its shaft, in one of which it is in gear, and in the other, out of gear. The latter is the arrangement more commonly met with. Where a turning engine is not provided, the turning wheel is usually arranged for operation by hand through worm gearing operated by a lever with pawl and ratchet arrangement, or by some similar device. In some cases the engine is turned by a hydraulic jack placed under a movable chock piece located in sockets cast in the turning wheel. This chock is shifted from one socket to another as the jack shoves it upward, and thus the engine is slowly turned. In small engines the turning wheel is often simply a form of gear wheel with shallow teeth in which a pinch bar is worked, and by this means the engine may be slowly pried around. Such a wheel is known as a ***pinch wheel.***

and both floating in a square steam tight box or cylinder, and being guided in their movements by the crank on the end of the driven shaft. In construction the inner piston has removable shoes which can be adjusted to compensate for wear, when necessary, and the cylinder is also provided with adjustable wedge for the same purpose.

FIG. 1,283.—Dake throttling square piston engine. Sizes range from 2 to 30 horse power, 800 to 250 revolutions per minute respectively. *In starting*, apply steam for a few minutes before commencing work. As soon as the metal is warm the engine will start readily. Do not pry or try to force the piston over till this is done.

The inner piston or cross head has cored through its body four ports leading to the four sides, which communicate with ports

cut in cover or cylinder head of engine. This engine is suitable for operating blowers, ventilating fans, centrifugal pumps or other small power applications where high speed operation is required, the small number of reciprocating parts adapting it to this service.

Rocking Valve Engine.—This type of engine is intended to meet the demand for an engine, where heavy duty is required, and where economy

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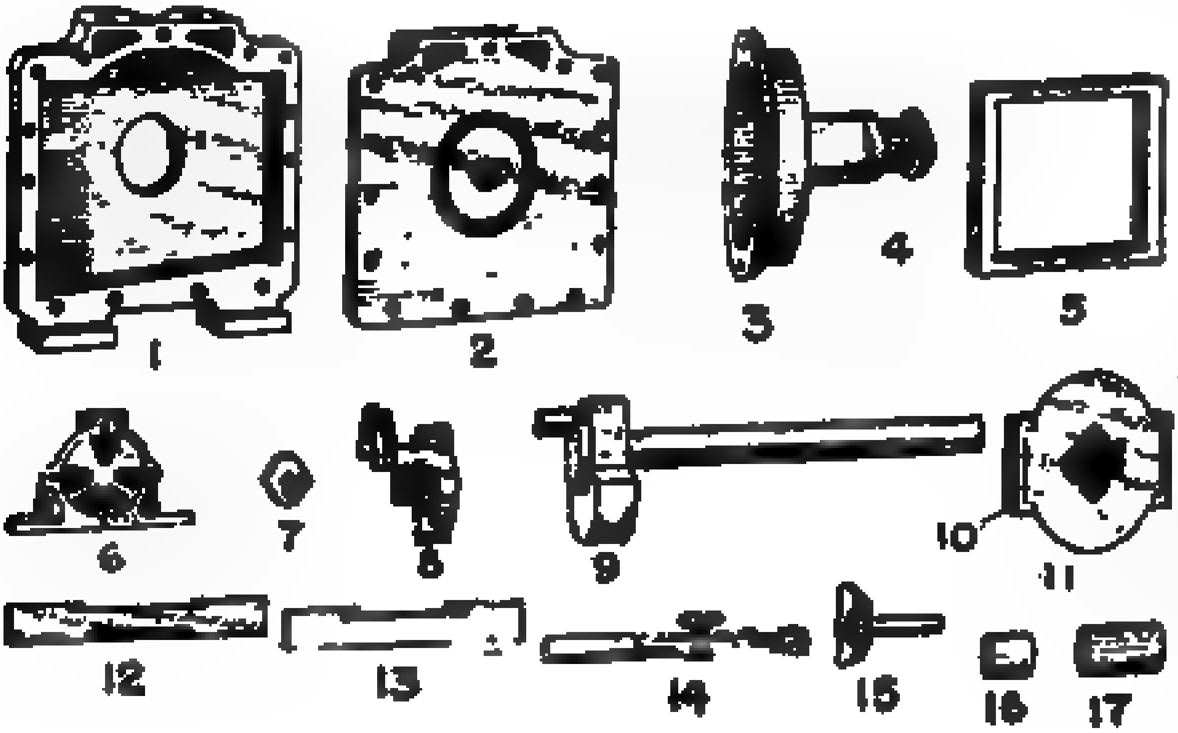


Figs. 1,287 to 1,301.—Parts of Dake square piston engine. 1, cylinder; 2, cover or cylinder head; 3, back horn or main journal; 4, gland; 5, outer piston; 6, valve chest; 7, valve stem gland; 8, valve cover; 9, crank and shaft; 10, piston slipper or gib, 11, inner piston; 12, slipper; 13, wedge; 14, valve lever; 15, valve and stem, 16, bushing of inner piston; 17, bushing of crank shaft.

Figs. 1,302 and 1,303.—Sectional views of Colt disc engine. ***In construction,*** the main body of the engine consists of one casting, containing six cylinders, arranged in a circle, and parallel with one another. The pistons A, take the form of a hollow plunger, one end terminating in a blunt cone which bears continuously against the periphery of the disc B. They are single acting, being subject to steam pressure upon the flat end only. Steam is admitted successively to the six cylinders from the steam chest C, three pistons being constantly in action at different points of the stroke, thereby imparting a uniform rolling motion to the conical disc B, which is steadied at its center by the ball and socket joint D, and rolls upon the conical surface of the back plate E, which receives the full thrust of the pistons, and protects the ball and socket joint D, from strain. The crank pin F, is securely fixed in the center of the conical disc B, the rolling motion of the disc causing the pin to describe a circle, and by means of the crank G, imparting a rotary motion to the shaft H. The shaft H, passes through the center of the steam chest and carries an eccentric giving motion to the circular valve K. The valve K, is a flat circular ring which slides steam tight but perfectly freely between the port face and a balance plate. The steam is admitted to and fills the annular space C, left in the steam chest outside the circumference of the valve ring K, the eccentric motion of which alternately opens and closes all the steam ports, successively admitting steam to the cylinders, from which it again escapes to the exhaust chamber M, formed by the inside of the valve ring, and thence through openings into the body of the engine, and is finally discharged by the exhaust pipe N.

valve engine has long been recognized as one especially adapted for saw mills, planing mills, etc., where a simple engine and one capable of furnishing great power is desired. The design of such parts of the frame, outboard bearing, connecting rod,

FIG. 1,304.—Southern rocking valve engine for heavy duty as in saw mills, planing mills, etc.

cross head, piston, etc., are of ordinary heavy duty form. The cylinders are provided with steam and exhaust ports of large areas.

The rocking valve, which is similar to a Corliss valve, has large wearing surfaces. The cylinder heads and valve bonnets are made tight by ground joints. Fig. 1,304 shows the general appearance of a rocking valve engine.

CHAPTER 22

AIR COMPRESSORS

The ever broadening field for the use of compressed air, and the rapid increase of invention of compressed air appliances, have produced a number of changes in the design of the earlier compressors, the scope of which was confined almost entirely to mine and tunnel work.

Compressed air is used in almost every art known to man, and in many cases its use depends so much upon its economical production, that the modern compressor must embody every refinement which has proved to be of practical value.

The Compression of Air.—When the space occupied by a given volume of air is changed, both its pressure and temperature are changed in accordance with the following laws:

Boyle's law: *At constant temperature, the* **absolute** *pressure of a gas varies* **inversely** *as its volume.*

Charles' law: *At constant pressure, the volume of a gas is proportional to its* **absolute** *temperature.*

In the ordinary process of air compression, therefore, two elements are at work toward the production of a higher pressure:

1. The reduction of volume by the advancing piston;

2. The increasing temperature due to the increasing pressure corresponding to the reduced volume.

The application of the two laws is illustrated in fig. 1,305, which shows a cylinder fitted with an air tight piston. If the cylinder be filled with air at atmospheric pressure (14.7 lbs. per sq. in. **absolute**) represented by volume A, and the piston

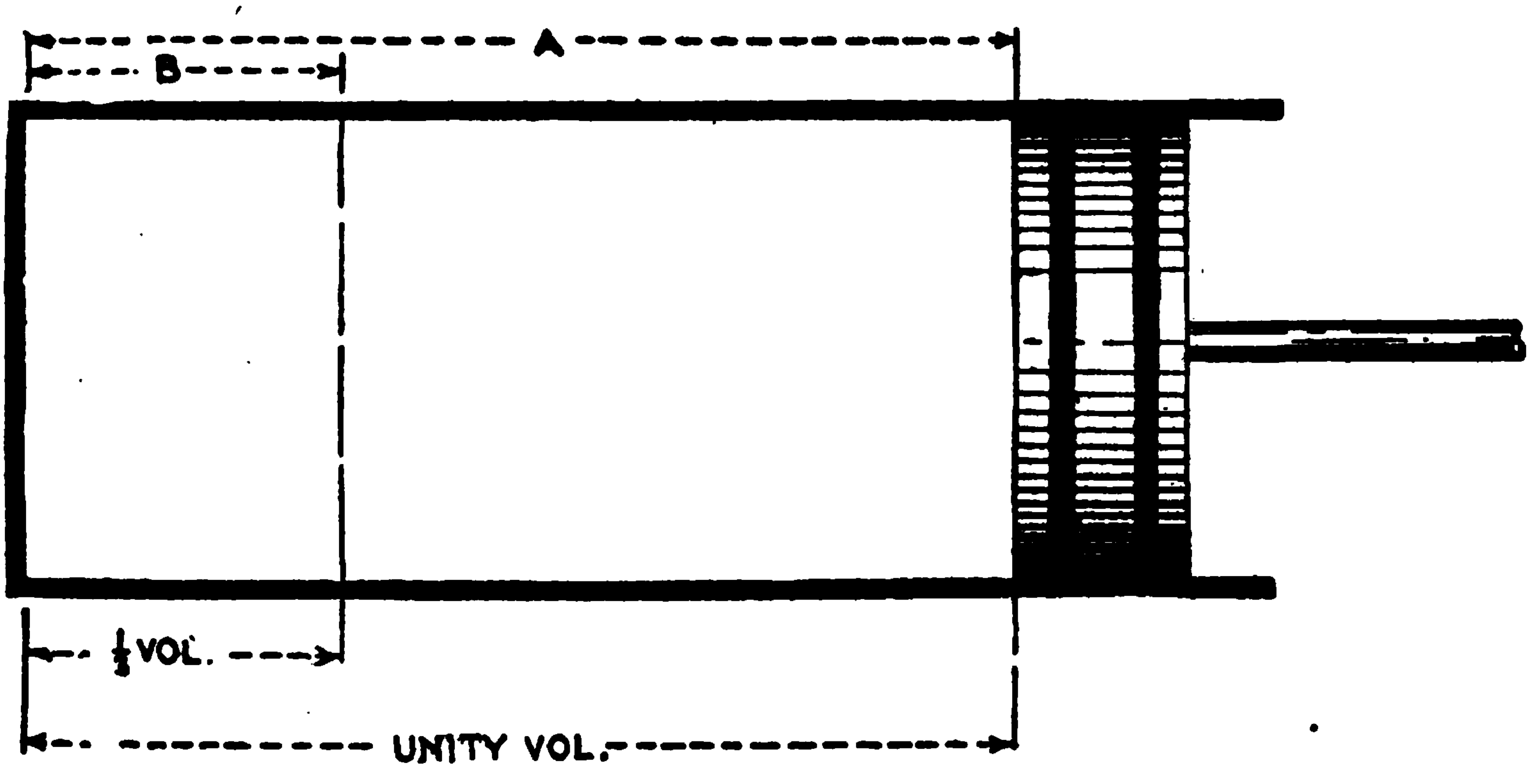


FIG. 1,305.—Elementary air compressor illustrating the phenomena of compression as stated in Boyle's and Charles' laws.

be moved to reduce the volume, say to $\frac{1}{3}$ A, as represented by B, then according to Boyle's law the pressure will be trebled or $= 14.7 \times 3 = 44.1$ lbs. absolute, or $44.1 - 14.7 = 29.4$ **gauge** pressure. ***In reality,*** *however, a pressure gauge on the cylinder would at this time show a higher pressure than* 14.7 **gauge** *pressure because of the* **increase in temperature** *produced in compressing the air.*

Now, in the actual work of compressing air, it should be carefully noted that *the extra work which must be expended to overcome the excess pressure due to rise of temperature* **is lost,** *because after*

the compressed air leaves the cylinder **it cools,** *and the* **pressure drops** *to what it would have been if compressed at constant temperature.*

Accordingly, in the construction of air compressors, where working efficiency is considered, some means of cooling the cylinder is provided, such as projecting fins. or jackets for the circulation of cooling water.

FIG. 1,306.—Sullivan steam driven, direct connected single stage air compressor on sub-base.

Ques. What is "free air"?

Ans. Air at ordinary atmospheric pressure and temperature, whatever these may be.

NOTE.—In air compressor problems careful distinction should be made between *gauge pressure* and *absolute pressure,* the former being the pressure as indicated by a pressure gauge, as distinguished from absolute pressure which is the gauge pressure plus 14.73 lbs., the weight of the atmosphere at sea level, when the barometer reads 30 ins. or, for ordinary calculations, 14.7 lbs.

FIG. 1,207.—***Air compression characteristics;*** curves showing the thermal result of air compression and expansion. The simplest application of this diagram is that which gives the gauge pressure represented at different points of the stroke. This is shown in the vertical lines. But in compressing air, heat is produced, and it is important to know the temperature at any given pressure, also the relative volume. All of these are shown in the diagram. The initial volume of air equal to one is taken and divided into ten equal parts. Each division between two horizontal lines, shown by the figures at the right, representing one-tenth of the original volume. The horizontal and vertical lines are the measures of volumes, pressures and temperatures. The figures at the top indicate pressure in atmospheres above a vacuum; the corresponding figures at the bottom denote pressures by the gauge. At the right are volumes from one tenth to one; at the left, degrees of temperature from zero to 1,000 degrees Fahr. The two curves which begin at the upper left hand corner and extend to the lower right are the compression curves. The upper one is the *adiabatic* curve, or that which represents the pressure at any point on the stroke, with the heat developed by compression remaining in the air; the lower is the *isothermal*, or the pressure curve uninfluenced by heat. The three curves which begin at the lower left hand corner and rise to the right are heat curves, and represent the increase of temperature corresponding with different pressures and volumes, assuming in one case that the temperature of the air before admission to the compressor is zero, in another 60 degrees, and in another 100 degrees. Beginning with the adiabatic curve, it will be noted that for one volume of air, when compressed without cooling, the curve intersects the first vertical line at a point between .6 and .7 volume, the gauge pressure being 14 7 pounds. If it be assumed that this air was admitted to the compressor at a temperature of zero, it will reach about 100° when the gauge pressure is 14.7 pounds. If the air had been admitted to the compressor at 60°, it would register about 176° at 14.7 pounds gauge pressure. If the air were 100° before compression, it would go up to about 230° at this pressure. Following this adiabatic curve until it intersects line No. 5, representing a pressure of five atmospheres above a vacuum (58.8 pounds gauge pressure), the total increase of temperature on the zero heat curve is about 270°; for the 60° curve it is about 370°, and for the 100° curve it is 435°. ***The diagram shows that*** when a volume of air is compressed adiabatically to 21 atmospheres (294 pounds gauge pressure), it will occupy a volume a little more than one tenth; the total increase of temperature with an initial temperature of zero is about 650°; with 60° initial temperature it is 800° and with 100° initial it is 900°. It will be observed that the zero heat curve is flatter than the others, indicating that when free air is admitted to a compressor cold, the relative increase of temperature is less than when the air is hot. This points to the importance of low initial temperature. It is plain that a high initial temperature means a higher temperature throughout the stroke of a compressor. ***The diagram gives*** the loss of temperature during compression from initial temperatures of 0°, 60°, 100°. Comparing the compression curve from zero with the compression curve from 100°, it will be noted that in compressing the air from, say, 1 atmosphere to 10 atmospheres, the original difference, which at the start was only 100°, has now been about doubled; that is, it has reached 200°, and in carrying

The Heat of Compression.—This subject has probably received more consideration in air compressor design than any other. The principal losses in the earlier compressors were traceable to this source.

FIGS. 1,308 to 1,315.—Parts of Ingersoll-Rand's *Corliss type inlet valve*. The working pressure is distributed over the entire valve surface, which is almost a half circle. *In operation*, ample port opening is provided at the beginning of the stroke, when the piston is moving most slowly. This opening increases toward mid-stroke; with the piston at its highest speed the port is fully open. The closing of the valve is timed to coincide with the stopping of the piston at the end of the stroke. Thus air is admitted for full stroke and shut off suddenly at the end, so that there can be no escape of free air.

FIG. 1,307.—*Text Continued.*
the compression to 20 atmospheres, the difference now becomes about 250°. Each vertical division represented by the figures at the left is equal to 100°, and the space between any two adjacent vertical lines may be subdivided into 100 equal parts representing 1° each. Where there is a system of cooling the air during compression, the lines on the indicator cards can be traced between the adiabatic and isothermal curves on the diagram. In practice, the best compressors show a line about midway between these two curves. For all practical purposes, in using the diagram it is best to follow the adiabatic curve in all determinations, except where the exact pressure line is known. This diagram will be found convenient to those who are called upon to figure the pressure at different points in the stroke of an air compressor and it points out the common error of neglecting to take into consideration in calculating, the fact that, at the beginning of the stroke, one atmosphere in volume already exists. Beginning at the upper left hand corner, the adiabatic pressure curve intersects the first vertical line at that point in the stroke, when the pressure on the gauge will register 14 7 pounds. The next vertical line shows where the gauge reaches 29.4 pounds, and it is evident here that the piston of an air compressor travels much farther in reaching 14.7 pounds than in doubling that pressure or in reaching 29.4 pounds; thus an air compressor is an engine of unevenly distributed resistance. During the early stages of the stroke it has a slowly accumulating load to carry, while later on this load is multiplied very rapidly. This is one of the reasons for heavy flywheels in air compressors.

By reference to a table giving the temperatures at end of compression for various terminal pressures, it is seen that aside from the injurious effect such high temperatures would have on the lubrication of the cylinder and valves of an air

FIGS. 1,316 to 1,320.—Parts of Ingersoll-Rand Imperial *direct lift discharge valve* and assembly. The valve proper is machined from steel and ground to seat; the projecting lip or rim above the seat is caught in the back lash of the compressed air when the piston stops and assists the spring in closing the seat quickly. The valve is cup shaped.

FIGS. 1,321 to 1,323.—Ingersoll-Rand *direct lift inlet valve*, with its bonnet and locking nut. Each unit is self-contained, screwing into place in the cylinder and locked with an auxiliary nut. The valve seats in the cage so that its condition is easily ascertained on the removal of a unit.

compressor, the thermal loss would grow as the pressure increased, if no means were provided for abstracting this heat during compression. It should be noted that the heat of compression, as already explained, represents work done upon the air for which there is

usually no equivalent obtained, since the heat is all lost by radiation, before the air is used.

Simple Compression.—In the earlier compressors, compression was accomplished in one stage or single cylinder machines, and the heat of compression was removed by injecting water into the cylinder in the form of a spray; or, in another type, the water was used as a piston for compressing.

FIG. 1394.—Sullivan small steam driven single stage air compressor. It is of the center crank pattern, with the main bearings, crank shaft and fly wheels at one end of the machine, and the steam and air cylinders both at the other or rear end. A cylindrical tie piece joins the steam and air cylinders. The compressor is supported on a sub-base, rendering it self-contained. The air valves consist of automatic poppet inlet and discharge valves. The valves are set close to the ends of the cylinder, the inlet valves at the bottom and the discharge valves at the top of the cylinder. These valves act in a radial direction to the cylinder axis thus reducing clearance. The frame is a solid casting, provided throughout its length with a broad foot and resting on a heavy sub-base. The rear end is flanged for the attachment of the team cylinder and the front end has two bearings, one on either side, to support the driving shaft. The central portion is bored cylindrically to serve as a guide for the cross head. A sheet steel guard surrounds the crank, to prevent oil being thrown upon the floor.

The spray injection cylinder has now given way almost entirely in this country to the dry or jacketed cylinder.

Ques. What are the features of the spray injection process?

Ans. A higher thermal efficiency may be attained than by the dry process, but from a commercial point of view its

efficiency is not so high, for the water in the cylinder prevents proper lubrication.

Impurities in the water also attack the walls of the cylinder, calling for repeated reboring of cylinder and other heavy repairs; the heat absorbs considerable of the moisture and this is deposited in the delivery pipe where it freezes in cold weather and restricts the passage of air, or it is carried to the motor, where it chokes the exhaust ports by reason of low temperatures resulting from expansion.

Ques. What are the features of the dry or jacketed compressor process?

Ans. In this system the external walls of the cylinder are flooded with cooling water which keeps the cylinder walls sufficiently cool so that proper lubrication is not interfered with and all other disadvantages of the wet compressor are obviated.

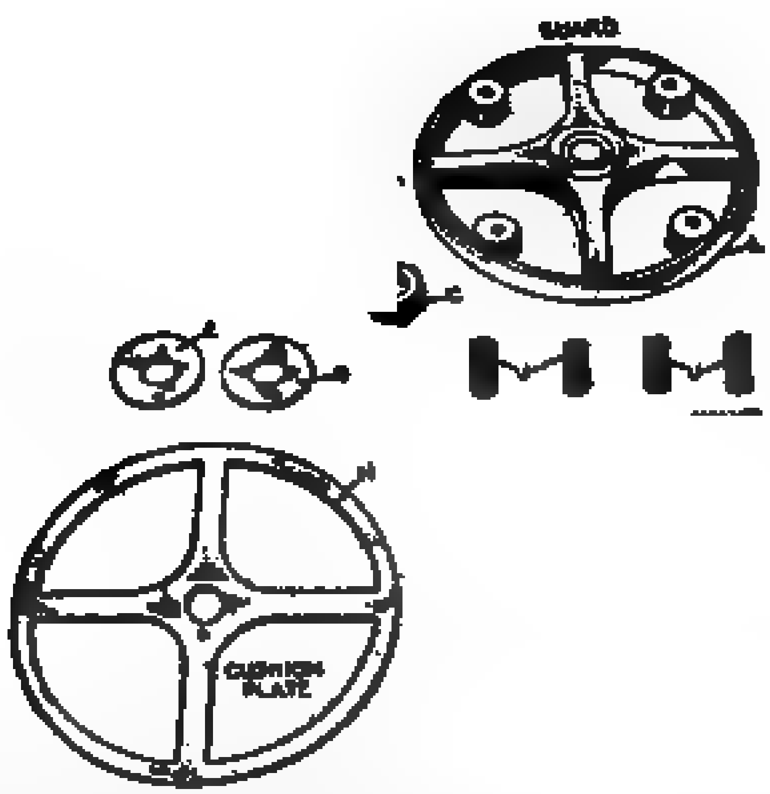
Compound Compression.—The efficiency due to the heat of compression decreases as the terminal pressure increases, and for pressures above 60 lbs., the water jacket of the simple compressor is not sufficiently effective for producing the most economical results, and stage or compound compression is resorted to as the most practical and efficient method for reducing the loss due to the *heat of compression.*

Ques. How are the cylinder diameters of a compound compressor proportioned?

Ans. They are so proportioned as to divide the work of compression equally between the given number of cylinders.

Ques. Describe the cycle of compound compression with intercooling.

Ans. Free air is admitted to the low pressure cylinder, where it is partially compressed, and then forced into an *intercooler.* The intercooler acts as a receiver and at the same time removes the heat of compression of the intake cylinder before the air



Figs. 1,325 to 1,336.—Cross section of Ingersoll-Rogler valve and parts. ***In construction*** **A is the valve seat; B, the valve bolt; C, the valve bolt nut; E, a washer placed between the valve seat and the valve proper. F is the valve; G a washer placed between the valve and cushion plate H. L is the guard; J, the valve springs, placed inside the four pockets of the guard and which act on the valve. Washers E and G, valve F, and stop plate H are clamped by means of valve bolt B and kept from turning by a dowel pin D. The portions M of the valve F are integral spring arms. They are ground to about half the thickness of the valve proper, and are made narrow, giving elasticity. The portions M of the valve F are integral spring arms. They are ground to about half the thickness of the valve proper and are made narrow, giving them great elasticity. These portions M should not be confused with the term springs; they are merely connecting arms between the fixed and the moving sections of the valve and serve to hold the valve in one position and seat it always in the same place. With valve at rest, it is held in its seat by the four main springs J, against a slight tension of the integral valve arms M. As soon as the air pressure required to open is reached, the valve opens against these four coil springs to very nearly its full opening. It then comes in contact with the cushion plate H and moves the last $\frac{1}{16}$ inch to $\frac{1}{64}$ inch of its travel against its additional spring tension, the cushion plate having a certain amount of elasticity, it being fixed in the center only. When the piston passes the dead center on the return stroke, the valve closes. The function of the cushion plate is to act as a buffer, absorbing any shock that might otherwise fall on the valve, thus prolonging the life of the valve and reducing the noise.**

FIG. 1,338.—Sectional view of steam end of Sullivan two stage air compressor. The steam distribution is regulated by a Meyer valve gear giving variable cut off by adjustable lap of the riding valve. The frame is a heavy box shaped casting strongly ribbed and extending the full length of the compressor. A solid bottom is provided under the steam end for collecting oil and drippings from the steam cylinder cross head guides and steam valve gear.

FIG. 1,339.—Sectional view of air end of Sullivan two stage air compressor showing Corliss type air valves, intercooler, etc. The inlet valves on both the high and low pressure air cylinders are of the rotary type and receive their motion from an eccentric pin located in the end of an overhanging arm on the crank pin. The motion from this pin is transmitted through a series of adjustable connecting rods and levers to the valve stems. The poppet discharge valves are of cup shaped form, and are internally guided on an extension of the valve plug with the springs inside, so that the springs cannot become clogged by accumulation of oil and dirt.

is admitted to the second stage cylinder. In the high pressure cylinder the process of compression is completed and the air is delivered to the receiver at the required terminal pressure.

The final temperature in each cylinder will be the same if the work has been divided equally and the intercooler properly designed, but it will be very much lower than if the compression were done in one cylinder. For instance, in compressing air to 100 pounds pressure in a two stage compressor, the air is compressed from atmospheric pressure to, say, 26½ pounds in the intake cylinder and is delivered to the

pressor. Probably the largest demand for high air pressures comes from the mining and contracting fields where pneumatic haulage is employed. Scientific and experimental work calls for still higher pressures in air and gas and the U.S. Government uses high pressure air for a variety of purpose.

intercooler at this pressure and at 240° Fahr.* (atmospheric temperature at 60° Fahr.). If all of the heat of compression be taken out by the intercooler, it is admitted to the high pressure cylinder at atmospheric temperature and is there compressed from 26½ pounds to 100 pounds and delivered to the receiver at 240° Fahr.*

In a single stage compressor the air is compressed from atmospheric pressure to 100 pounds in one cylinder and reaches the receiver at 482° Fahr.*

* NOTE.—Radiation and cooling influence of water jacket not considered.

FIG. 1341.—**Compression curves.** **If the heat of compression be removed as fast as generated,** and the compression proceed under a uniform temperature, it is said to be *isothermal,* and the compression curve follows the line A B C in the diagram. If the isothermal compression of air were practicable, and if the heat absorbed from the air by the cooling agent could be applied to the air during its re-expansion in use along a similar isothermal curve, the result would be perfect thermo-dynamic efficiency, and it is toward an approximate realization of these ideal conditions that the best compressor practice is directed. The line A D E in the diagram represents the other extreme, or *adiabatic* compression, during which no heat is removed from the air, and in consequence of which the temperature rises. Since a given weight of air at a given pressure occupies a volume proportional to its absolute temperature, the volumes at all pressures are greater in adiabatic than in isothermal compression in a ratio increasing with the pressure, and since the work of compression and discharge is represented by the area in the diagram to the right of the curve along which compression is accomplished, it will be seen that the area A D E C B stands for the waste of energy caused by allowing the heat of compression to remain in the air. ***The problem of economy,*** therefore, becomes one of the abstraction of the heat generated in the air during the process of compression; and the most obvious solution lies in keeping the air during that period in intimate contact with cold water. ***In the early days*** of slow running machines the injection of a water spray into the cylinder was attended with success, but the danger and impracticability of such a method with modern speeds must be apparent, and have in fact caused the direct spray to be generally discarded. As a partial substitute for the injection of water, good practice requires the ***water jacketing*** of the air cylinder, but owing to the short interval within which compression takes place, there is very little actual cooling of the air therein, and the jackets are chiefly useful in keeping the cylinder walls cool enough for effective lubrication, and in the prevention of cumulative heating. At ordinary speeds, the indicator card from an air cylinder in good condition, shows a compression line approaching the adiabatic curve A D E much more closely than the isothermal A B C, so closely indeed, that within the usual and proper ranges of single cylinder compression, no great error is introduced by regarding the compression as adiabatic. Any claims as to approximately isothermal single stage compression must attract suspicion whether supported by cards or not, since such cards can be based only on abnormally slow running, a spray injection, or most probably on leaky suction valves or piston. The loss of energy in adiabatic (approximately single stage) compression is represented by the area A D E C B. The saving effected by two stage compression is, therefore, represented by the unshaded portion D E H B. It is evident that this unshaded area, representing the two stage saving, will increase or decrease with the terminal pressure.

FIG. 1,342.—Eccentric pin for driving air valve gears of Sullivan Corliss valve type, air compressor. The pin is located in the end of an overhanging arm on the crank pin as shown, motion from this pin being transmitted through a series of adjustable connecting rods and levers to the valve arms. The valve gear is so arranged that the valves, during their opening and closing movements, at the end of the stroke of the pistons, are moving at their greatest speed, but at the time when they are fully open, at mid-stroke, during which the piston is traveling at its highest speed, they remain at rest or nearly so, so that the cylinder will be filled with air at practically the full barometric pressure.

FIG. 1,343.—Front end of Sullivan "straight line" riding cut off air compressor showing steam valve gear. There are two fly wheels, one on each end of the shaft as shown. The balanced slide valves are operated by two eccentrics on the crank shaft between the main bearings through two upright rockers seen at the end of the bed plate.

Ques. What is the principal advantage of compound compression over simple compression?

Ans. The reduction of the loss due to the heat of compression.

Other important advantages due to compounding may be enumerated as follows:

FIG. 1,344.—Rand air cylinder showing control devices. *Unloading device:* The speed of a belt driven air compressor cannot be controlled the same as that of a steam driven machine; its regulation must be accomplished either by throwing off the load, or by stopping the machine during the intervals when there is no demand for compressed air. Accordingly on belt driven compressors a device is provided called an *unloader*, which is placed in the air inlet pipe close to the intake cylinder, and unloads the compressor by cutting off the supply of air. When the unloader is in action, no work is done by the machine excepting that necessary to overcome friction. *Relief starting valve:* This device is used on compressors employing poppet inlet valves. One is placed in each head and consists of a screw, with a knurled handle, threaded into an inlet valve bonnet, and when screwed in, it forces the inlet valve from its seat. By thus holding an inlet valve open in either head, the compressor may be started without load, and when full speed is attained, the valves may be closed, and the compressor permitted to perform its regular work.

1. Cooler intake air;
2. Better lubrication;
3. Reduction of clearance losses;
4. Lower maximum strains and nearer uniform resistance.

The temperature of air leaving the intake cylinder being low, the cooling influence of the jacket is better, the cylinder walls are cooler

between strokes, and the air enters the cylinder cooler than in a single stage compressor. The lubricant for cylinders and valves is not subject to the pernicious influence of high temperatures; and the clearance losses, or losses due to dead spaces, are less in a compound compressor than in a simple compressor. Clearance loss in an air compressor is principally a loss in capacity, and therefore affects only the intake cylinder; it increases with the terminal pressure, but since the terminal

FIG. 1,346.—Ingersoll-Rand Imperial unloader. *It operates* by closing the air intake, and automatically regulates the compressor, maintaining the line pressure within a satisfactory range. *It consists* of a valve on the compressor intake pipe, normally held open by its own weight. When receiver pressure rises above normal this pressure is communicated through a small pipe from the regulating valve, acting to close the valve in the intake pipe, thus shutting off the air supply. The regulating valve can be adjusted for any working pressure by means of a spring adjusting screw.

NOTE.—Air compressor builders and those who use compressed air will agree that the problem of heating or cooling air is a difficult one. Hot air in the cylinder of an air compressor means a reduction in the efficiency of the machine. The trouble is, that there is not sufficient time during the stroke to cool thoroughly by any available means. Water jacketing is the generally accepted practice, but it does not by any means effect thorough cooling. The air in the cylinder is so large in volume that but a fraction of its surface is brought in contact with the jacketed parts. Air is a bad conductor of heat and takes time to change its temperature. The piston while pushing the air toward the head, rapidly drives it away from the jacketed surfaces so that little or no cooling takes place. This is especially true of large cylinders, where the economy effected by water jackets is considerably less than in small cylinders. Engineers who are shown indicator cards from large air compressors with pressure lines running away from the adiabatic, naturally regard them with suspicion, and look for leaks past the piston or through the valves. Such leaks will explain many isothermal cards, and until something better than a water jacket is devised, it is well to seek economy in air compression through compounding. The great advantage of compounding is in the fact that more time is taken to compress a certain volume of air, and that this air while being compressed is brought into contact with a larger percentage of jacketed surfaces.

NOTE.—Since the power driven compressor is almost always a constant speed machine various methods of regulation are employed. Constant speed means constant piston displacement; the problem of delivering a variable volume of air with constant piston displacement, becomes one of making a portion of that displacement non-effective in the compression and delivery of air. Only the fundamental principles of several methods of accomplishing this will here be discussed.

NOTE.—***The first method*** is really one of unloading, rather than of regulating. *A pressure controlled mechanism is arranged so that when pressure exceeds normal a communication is opened between the two sides of the compressing piston.* Usually this is accomplished by opening and holding open one or several of the discharge valves at both ends of the cylinder, the air is then simply swept back and forth from one side of the piston to the other through the open valves and the air discharge passage. When normal pressure is restored, the valves are automatically closed, and compression and delivery are resumed. Evidently this is practically a total unloading of the machine for a longer or shorter period—a sudden release from load and a sudden resumption of load. Moreover, the air which is swept back and forth by the piston in its travel is air under full pressure; so that when the discharge valves suddenly close, the piston at once encounters a full cylinder of air at maximum pressure. These facts limit regulators of this class to machines of comparatively small capacity.

NOTE.—***The second method*** *provides, by means of a pressure operated device, for the partial or total closing of the compressor intake under reduced load.* To avoid the dangers attendant upon such an operation acting suddenly, these devices are provided with some damping mechanism so that they are compelled to operate slowly, making the release or resumption of the load gradual. The cutting down of the air intake results in a rarefication of the air entering the cylinder, and a greater range between initial and discharge pressures, with a corresponding increase in the range of temperatures. This method of regulation, therefore, is not suitable for very great load variations.

NOTE.—***The third method*** is very similar to the first, except *that here the inlet valves, instead of the discharge valves, are held open when the machine is unloaded,* the piston thus simply drawing in and forcing out air at atmospheric pressure. It is open to the same criticism (though in somewhat less degree) as the first method, namely, undue shock and strain on release and resumption of load.

NOTE.—***The fourth method*** *uses a pressure controlled valve on the compressor discharge of single stage machine, combining also the functions of a check valve to limit the escape of air from the receiver or air line.* Excessive pressure blows the discharge to atmosphere, instead of into the line. This arrangement is also used on two stage machines by placing it on the low pressure discharge to the intercooler. Then, when the governor valve is opened by excess pressure, the low pressure cylinder discharges to atmosphere, and the high pressure cylinder acts simply as a low pressure cylinder with intake at atmospheric pressure. This device is more of a relief valve than an unloader, for the piston must continue to compress to a pressure which will open the discharge valves; this volume of compressed air is wasted.

NOTE.—***The fifth method*** *provides auxiliary clearance spaces, or pockets, at each end of the cylinder, which are successively "cut in" as load diminishes.* The excess air is simply compressed into these clearance spaces and expanded on the back stroke. The capacity of the cylinder is reduced without any appreciable waste of power; for the energy used in compressing the clearance air is given back by its expansion.

NOTE.—On power driven ***compressors with Corliss intake valves,*** several different methods of unloading or regulating are used. ***By one method,*** *the Corliss valve is held open for the full admission stroke, and also for a part of the compression stroke,* this latter portion being determined by the unloading called for. Evidently this is practically equivalent to a shortening of the stroke of the compressor. ***By another method*** *the Corliss intake valve is opened full at beginning of admission, but closes later in the admission stroke.* The air admitted to that point is expanded or rarefied for the remainder of the compression stroke, and then compressed, the volume of compressed air delivered being of course reduced. This arrangement is productive of an excessive temperature range in the cylinder. Still a ***third method*** *opens and holds open the intake valves at the end of the cylinder, or at opposite ends in duplex machines.* The effect of this is to make ineffective one out of every two strokes. If still further unloading be necessary, the intake valves at the other end of the cylinder or cylinders are opened and held open. The three arrangements just outlined all operate by a pressure controlled mechanism which actuates some form of trip on the Corliss air valve gear.

NOTE.—Three things are to be avoided in the successful unloader or regulator for power driven machines; first, a sudden release or resumption of load, throwing heavy strains on the machine: second, undue rarefication of the intake air, resulting in a wide range of cylinder pressures and temperatures; third, the blowing off of compressed air to the atmosphere with a waste of power.

pressure of the intake or low pressure cylinder of a compound compressor is much less than the terminal pressure of a simple compressor, the volumetric efficiency of the compound compressor is greater than that of the simple compressor.

The life of a compound compressor is longer than that of a simple compressor for like duty, due to better distribution of pressures.

Intercoolers.—A properly designed intercooler should reduce the temperature of the air back to the original point, that is,

FIGS. 1,346 and 1,347.—Sullivan intercooler tubes and shell. The intercooler consists of a cast iron shell mounted on the air cylinders, and containing two nests or groups of copper or aluminum tubes through which the cooling water circulates. Suitable baffles, placed in the interior of the shell, break up the straight flow of air, causing it to come thoroughly in contact with the tubes, in order that as much heat may be taken away from it as possible before entering the high pressure cylinder. The intercooler shell is so arranged that by removing the heads, either of the two nests of tubes may be drawn from the intercooler for cleaning or repairs. The tubes are firmly expanded into the tube heads at each end, obviating the use of packing or gaskets. In order to take care of variation in the length of the tubes, due to contraction and expansion, the intercooler is arranged so that the outer tube head of each nest is firmly bolted to the end of the intercooler shell, while the other head is free to come and go.

NOTE. -It is usually desirable to start a power driven compressor with no load, throwing on the load gradually after normal speed has been reached. This is in fact essential in machines driven by electric motors, for the heavy inrush of current in starting under load is dangerous, particularly where power is taken from a transmission circuit supplying other motors. Evidently almost any of the unloading devices noted in the previous section can be used for this purpose if properly arranged for manipulation. The usual form, however, is simply a by pass valve to atmosphere on the line close to the compressor protected by a check valve between it and the receiver to prevent the return of air from the line when the starting unloader valve is open. This check valve is essential where several compressors serve one line; permitting cutting in or out any machine without unloading the others. This by pass valve is opened on starting, when the compressor simply compresses to a pressure sufficient to open its discharge valves, this air escaping to atmosphere. When normal speed is reached, the by pass or unloading valve is gradually closed and load resumed. On two stage machines, an unloader valve should be provided on the low pressure discharge to the intercooler, as well as on the high pressure discharge to the line. In the latter case, both cylinders operate momentarily as low pressure cylinders.

to the temperature of the intake air. It can even do more than this, especially in winter, when the water used in the intercooler is of low temperature. A simple coil of pipe submerged in water is not an effective intercooler, because the air passes through the coil too rapidly to be cooled to the core, and such intercoolers do not sufficiently split up the air to enable it to be cooled rapidly. This splitting up of air is an important point. A nest

FIG. 1,348.—Ingersoll-Rand horizontal aftercooler with brass tubes. *It consists of* a horizontal shell supported on cast foot pieces. Air inlet and discharge connections are usually at the top and the water drain at the bottom. Within the shell is a nest of tinned brass tubes expanded in steel tube plates at each end, an expansion joint being provided at one end. Baffle plates face a cross circulation of the air over the tubes. The cooling water enters the lower set of tube, traversing each row, forward and back, leaving at the top of the one end of the after cooler. *In another design* galvanized iron pipes are used instead of brass tubes. These pipes are of two sizes, the larger telescoping the smaller and so arranged in pairs that water flows *in* through the inner tube and *out* through the annular space between the inner and outer tubes.

NOTE.—H. V. Haight in *American Machinist* says: "In multi-stage air compressors, the efficiency is greater the more nearly the temperature of the air leaving the intercooler approaches that of the water entering it. The difference of these temperatures for given temperatures of the entering water and air is diminished by increasing the surface of the intercooler and thereby decreasing the ratio of the quantity of air cooled to the area of cooling surface. Numerous tests of intercoolers with different ratios of quantity of air to area of surface, on being plotted, approximate to a straight line diagram for which the following figures are taken:

Cu. ft. of free air per minute per sq. ft. of air cooling surface ..	5	10	15
Diff. of Temp. F. between water entering and air leaving.......	12.5°	25°	37.5°

of tube carrying water and arranged so that the air is forced between and around the tubes is an efficient form of intercooler. If the tubes be close enough together and are kept cold, the air must split up into thin sheets while passing through. Such devices are naturally expensive, but first cost is a small expense when compared with the efficiency of the compressor, measured in terms of coal and water consumed. Receiver intercoolers are more efficient than those of the common type because

FIG. 1,349.—Horizontal air receivers. The discharge from an air compressor is more or less pulsating in character, and the receiver is analogous to an electrical rectifier, that is it receives and absorbs the pulsations, delivering a steady flow to the pipe line. It is, in a very small degree, an accumulator in which excess energy is momentarily stored and withdrawn. The receiver should be placed as close as possible to the compressor or after-cooler, and it is good practice to make the pipe between the receiver and compressor a size larger than that leaving the receiver. A safety valve and pressure gauge should be provided, and when the receiver is out of doors, the safety valve should be piped back into the compressor room to avoid freezing. There is some cooling of the air in the receiver, and since cooling involves condensation, the receiver should preferably be placed out of doors, where its cooling effect will reduce the moisture in the air. A drain cock should be provided at the lowest point which should be opened often for the discharge of accumulated water. Primary or main receivers, or those next to the compressor, should be so piped up that the air will enter at the top and leave near, but a little above, the bottom. On secondary receivers this arrangement should be reversed. On long pipe line systems small receivers, or moisture traps, should be placed at the low points in the lines, the piping entering and leaving at the top. These will catch the moisture condensed in the lines, which should be withdrawn frequently through a drain cock.

NOTE.—The successful intercooler must provide for a complete and minute subdivision of the air passing through it, that the heat may be dissipated without any dependence upon the heat conducting qualities of the air itself. The air should be split up into thin sheets or streams so as to dissipate its internal heat. There must be an ample cooling surface presented to this subdivided air stream.

the air is given more time to pass through the cooling stages because of the freedom from wire drawing which may take place in intercoolers of small volumetric capacity.

After Coolers.—The function of an after cooler is *to reduce the temperature of the air after the final compression.* In doing this it serves as a drier, reducing the temperature of air to the dew point, thus abstracting moisture before the air is started on its journey. In cold weather, with the air pipes laid over the ground, an after cooler may prevent accumulation of frost in the interior

FIG. 1,350.—Ingersoll-Rand vertical type after cooler, half in section. *In construction,* it has a shell or body made of steel throughout, with the exception of the head, which is an iron casting. Tinned brass tubes expanded in steel tube plates stand vertically in the body with provision for expansion and contraction at the lower end. Water enters at the lower end of the nest of tube leaving at the top. The air enters at the top or head, and leaves at the bottom, surrounding the tubes in transit and traveling counter current to the water. An open funnel in the water discharge shows the flow of water and permits of its proper adjustment. The enlargement of the body at the bottom gives a little receiver capacity and also catches the condensed moisture which may be drained off at intervals. A plate in front of the air discharge, guards against the escape of water which may be splashed up by the flowing air.

NOTE.—The after cooler has been devised to perform the cooling and drying functions by bringing the hot moist air from the compressor discharge in contact with the water cooled surfaces of such extent and during such a time that the moisture in the air will be condensed and deposited before it can enter the distribution system. Obviously the proper proportioning of the cooling surface and air velocity to volume of air to secure complete after cooling is a rather complex problem which needs a wide experience for its best solution.

NOTE.—After coolers, with adequate cooling surface, and properly supplied with water, will readily reduce the temperature of the compressed air to within 15 or 20 degrees of that of the cooling water. Obviously, the cooler the water supplied, the more complete the cooling and drying effected. The following figures (according to Ingersoll-Rand) are based on good cooling results with air at 80 to 100 lbs. pressure: When the temperature of the cooling water is 50, 60, 70, 80, 90 degrees Fahr., the gallons per hour required per 100 cubic feet of free air per minute, are respectively 120, 140, 160, 180, 200.

walls of the pipes, for where the hot compressed air is allowed to cool gradually, the walls of the pipe in cold weather act like a surface condenser and moisture may be deposited on the inside, for the same reason that frost appears on the inner side of a window pane. Another advantage of the after cooler is that it keeps the temperature of the pipe uniform, otherwise this pipe will be hottest near the compressor, gradually cooling down and being thus subject to irregularities of expansion and contraction.

The Saving Due to Compounding.—The table here given will serve to illustrate the large saving that it is possible to

Work Lost in Terms of Isothermal and Adiabatic Compression

Gauge Pressures	One Stage: Percentage of work lost in terms of isothermal compression	One Stage: Percentage of work lost in terms of adiabatic compression	Two Stages: Percentage of work lost in terms of isothermal compression	Two Stages: Percentage of work lost in terms of adiabatic compression	Four Stages: Percentage of work lost in terms of isothermal compression	Four Stages: Percentage of work lost in terms of adiabatic compression
60	30.00	23.00	13.38	11.80	4.65	4.45
80	34.00	25.26	15.12	13.12	5.04	4.80
100	38.00	27.58	17.10	14.62	8.00	7.41
200	52.35	34.40	23.20	18.88	9.01	8.27
400	68.60	40.75	29.70	22.90	12.40	11.04
600	83.75	44.60	32.65	24.60	15.06	13.10
800	90.00	47.40	35.80	26.33	16.74	14.32
1000	96.80	49.20	39.00	28.10	16.90	14.45
1200	106.15	51.60	40.00	28.60	17.45	14.85
1400	108.00	52.00	41.60	29.40	17.70	15.00
1600	110.00	53.30	42.90	30.00	18.40	15.54
1800	116.80	54.00	44.40	30.60	19.12	16.05
2000	121.70	54.80	44.60	30.80	20.00	16.65

NOTE.—In the above table no account is taken of jacket cooling, it being a well known fact among pneumatic engineers that water jackets, especially cylinder jackets, though useful and perhaps indispensable, are not efficient in cooling, especially so in large compressors. The volume of air is so great in proportion to the surface exposed and the time of compression so short, that very little cooling takes place. Jacketed heads are useful auxiliaries in cooling, but it has become an accepted theory among engineers that compounding or stage compression is more fertile as a means of economy than any other system that has yet been

effect by compounding. This table gives the percentage of work lost by the heat of compression, taking isothermal compression, or compression without heat, as a base.

Altitude Compression.—The height of the atmosphere surrounding the earth has been variously estimated to extend from fifty miles to twenty thousand miles, and since air has weight it exerts a pressure upon surrounding objects which is equal to the weight of the air column above the object.

Since air is very elastic its weight will cause it to have a variable density throughout its height and exert varying pressures at different altitudes.

At the sea level an atmospheric column balances a column of mercury 30 inches high and of equal area, which corresponds to a pressure of 14.7 pounds per square inch. The variation in pressure for different elevations has been determined by barometric observations and by examining a table of such observations it will be noted that the atmospheric pressure decreases with increasing height, and as a consequence one pound of air occupies a greater volume at an altitude than at the sea level (at the same temperature); or a cubic foot of air weighs less at a higher altitude than at a lower one.

In descending the shaft of a mine the contrary effect is noticed, but in a mine or any level below the sea, increase in density is counterbalanced by increase in temperature as the center of the earth is approached. The temperature of the atmosphere also changes with increasing altitude, but is not always uniform for any two places at the same elevation.

NOTE.—*Continued from page 726.*
devised. The two and four stage figures, as given in this table, are based on reduction to atmospheric temperature 60° Fahr. between stages. This is an important condition, and in order to effect it, much depends on the intercooler. This device represents a case of jacket cooling which in practice has been found to be efficient where engineers specify intercoolers of proper design. While cooling between stages the air may be split up into thin layers and thus cool it efficiently in a short time, a condition not possible during compression. This splitting up process should be done thoroughly, and while it adds to the cost of the plans to provide efficient coolers, it pays in the end. A rule which might be observed to advantage among engineers is to specify that the manufacturers should supply a compressor with coolers provided with one square foot of tube cooling surface for every ten cubic feet of free air per minute furnished by the compressor at its normal speed. Referring again to the table, it will be noted that when air is compressed to 100 pounds pressure per square inch in a single stage compressor without cooling, the heat loss may be 38 per cent. The condition, of course, does not exist in practice, except, perhaps, at exceedingly high speeds, as there will be some absorption of heat by the exposed parts of the machine. It is safe, however, to say that in large air compressors that compress in a single stage up to 100 pounds gauge pressure, the heat loss is 30 per cent. This, as shown in the table, may be cut down more than one-half by compounding or compressing in two stages, and with four stages this loss is brought down to 8 per cent. theoretically, and perhaps to 3 or 5 per cent. in practice. As higher pressures are used, the gain by compounding is greater.

The volumetric efficiency of an air compressor, expressed in terms of *free air*, is the same at all altitudes (for the displacement in a given size of cylinder is the same); but the volumetric efficiency, expressed in terms of *compressed air* at a given pressure, decreases as the altitude increases, for the quantity of air taken into a given cylinder per stroke being less dense at an altitude (due to lower initial or atmospheric pressure) it must be compressed into a smaller space for a given terminal pressure.

Example.—300 cubic feet of air, at atmospheric pressure of 14.7 pounds, compressed to 80 pounds gauge, will represent a volume of $300 \times 14.7 \div 94.7 = 46.6$ cubic feet.

If the atmospheric pressure were 10.1 pounds in the above example, then the volume delivered would be $300 \times 10.1 \div 94.7 = 34.1$ cubic feet; or the volumetric efficiency of a compressor performing the above work at the lower initial pressure would be but 73.2 per cent. of what it would be at the higher initial pressure.

In order, therefore, that an air compressor may deliver, at an altitude corresponding to the lower initial pressure, a volume of compressed air per stroke equal to that which it would deliver at a level corresponding to the higher initial pressure, the corresponding intake cylinder must be proportionately larger for the lower initial pressure as compared with the higher initial pressure.

CHAPTER 23

BLOWING ENGINES

Blowing engines are used for compressing large quantities of air against comparatively low pressures, as in blast furnace work where 15 to 30 lbs. per square inch is common. They are also employed in supplying large amounts of air or gas under pressure for chemical works and smelting furnaces, and for pumping and compressing natural and artificial gases:

The size of blast furnaces has grown to such an extent that the blowing engines which they require have become machines of magnitude.

The blowing engine is almost identical with the air compressor, the chief differences being in size and ratio of steam cylinder to air cylinder. Since a blowing engine delivers a large volume of air at low pressure, and an air compressor furnishes a small volume at high pressure, the cylinder ratios are quite different.

Blowing engines are usually vertical. The advantage of this arrangement, aside from the small floor space, is the reduced wear of the piston and cylinder walls. In the vertical engine, the shaft is generally placed at the bottom, and the air cylinder at the top, the arrangement being known as the "long cross head" type.

The Air End.—This is usually called the "blowing tub" and consists of a large cylinder as shown in fig. 1,351, fitted with a piston and having inlet and outlet valves in each end. They are built without any provision for cooling the air during compression, as it is not necessary, owing to the moderate pressure pumped against.

Formerly the old fashioned leather flap valve was universally used, followed by various modifications of the mushroom valve, which was moved in one direction by the air pressure, and in the other by a spring. To protect them from undue jar, various forms of cushion were used. All these required frequent renewal.

Owing to the great cost of maintaining blowing engines for Bessimer, and blast furnace purposes, due to the continued crystallization and breakage of the ordinary metal valve, and the liability to accident from such breakages, engineers have given much thought to the design of some

FIG. 1,351.—Southwark horizontal "blowing tub," with mechanically operated gridiron valves. View showing valve gear, and relief valves in the head.

device to reduce the repairs and liability to accidents, and at the same time increase the efficiency of engines of this class.

Formerly about 400 feet was the usual piston speed but with modern mechanically operated air valves this speed may be considered increased.

Fig. 1,352 is a view of the head of the tub shown in fig. 1,351.

It is fitted with gridiron air valves. They are rectangular in shape and extend entirely across the cylinder, thus giving a large inlet and outlet area. Each head of the cylinder contains one inlet and one outlet valve. The advantage is here obtained of the characteristic feature of the gridiron valve of giving a large port opening with small valve movements.

Fig. 1,352.—Southwark horizontal blowing tub. Sectional view showing valve mechanism. The inlet valve is operated positively by cam rollers. Movement of the outlet valve is obtained by a small cylinder having a piston attached to the valve stem. The cylinder being in communication with the tub, when the pressure builds up in the latter, the piston in the cylinder is pushed outward and the valve opens, being cushioned by a dashpot.

The valves bear against their seats in the direction of the air pressure.

The inlet valves are positive in their movement, each being actuated by a cam, which derives its motion from a convenient point on the valve gear.

The outlet valves are automatic in their opening action, being operated by a small auxiliary cylinder, the piston of which is attached to the valve stem; this is in communication with the air compressing cylinder and

provision is made so that when the pressure begins to build up, the piston in this small cylinder is pushed outward, and the valve opens. Any slamming of the valve is prevented by a dash pot. The closing is brought about mechanically by the action of a cam.

When the pressure in the blowing cylinder equals that in the receiver, air admitted from the blowing cylinder to the auxiliary cylinder causes the outlet valve to open. At the proper time a cam driven from the main

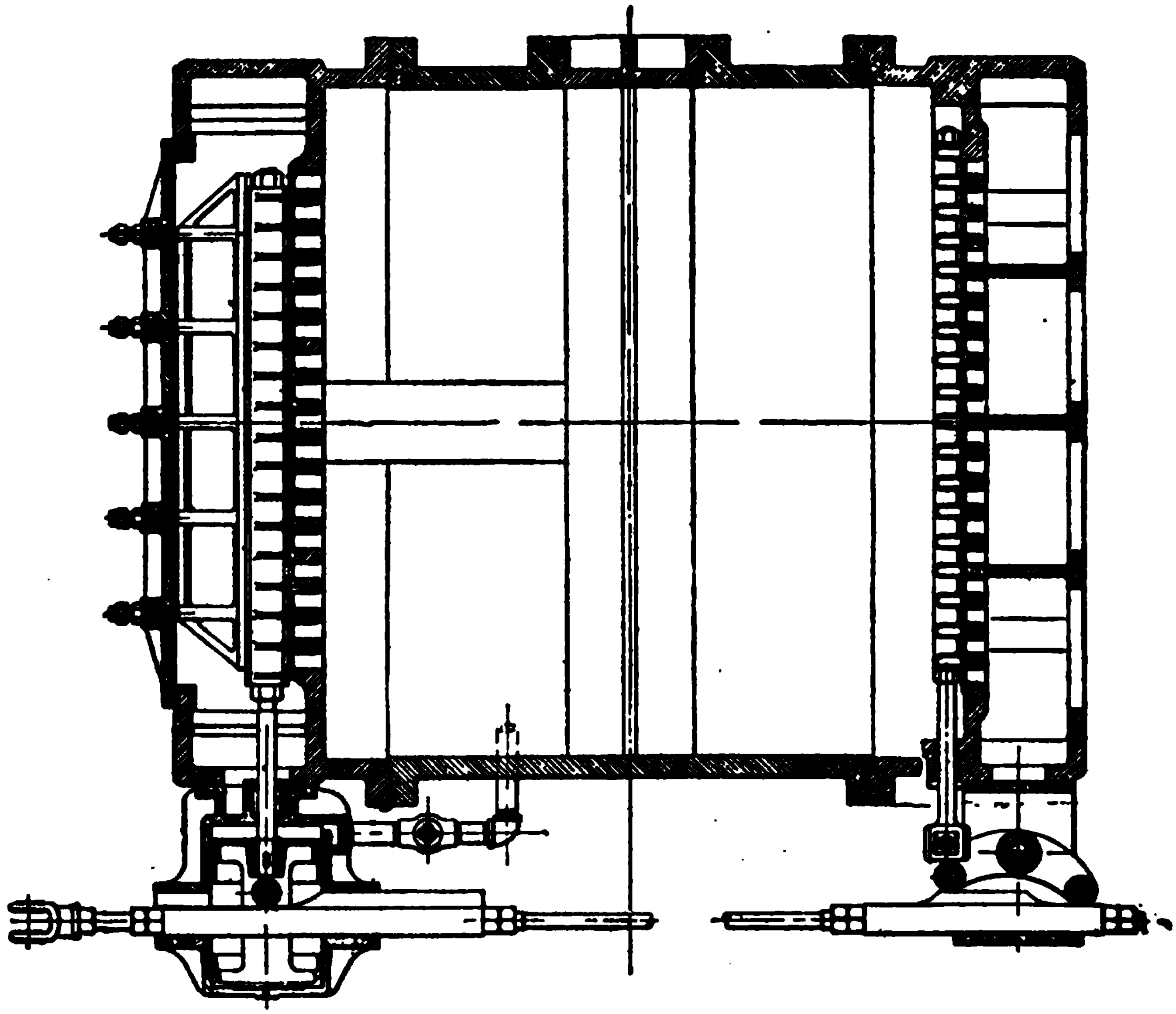


FIG. 1,353.—Southwark air cylinder head, inside view, showing inlet valve on the inside and seat of outlet valve above.

valve gear closes the valve by means of a rocker lever attached to the valve stem.

The Steam End.—With the advance of engineering knowledge, the demand has steadily increased for a higher efficiency in blowing engines. To meet this requirement the steam cylinder is fitted with the Corliss valve gear, and steam expanded to the degrees of maximum economy.

FIGS. 1,354 to 1,358.—Various types of ***Southwark blowing engines***. They are used for supplying large quantities of air under low pressure; in blast furnace practice, 15 to 30 lbs. is common, and in Bessemer steel works, 25 to 30 lbs. They are also used for chemical works and smelting furnaces, and for pumping and compressing natural and artificial gases.

Vertical long cross head type. Steam valves controlled either by hand or by governor.

Vertical compound steeple quarter crank type. The air cylinders are above the steam cylinders.

Horizontal cross compound quarter crank type with air cylinder tandem to steam cylinder.

Vertical compound quarter crank disconnected type. With the quarter crank arrangement of air and steam cylinders, the maximum pressure upon the steam piston before cut off is balanced by resistance upon the air piston.

Vertical compound quarter crank disconnected open frame type. The two inner cylinders are the steam cylinders, and the two outer, the air cylinders.

FIG. 1,361.—Southwark horizontal cross compound blowing engine disconnected.

On the long cross head type of blowing engine as shown in figs. 1,359 and 1,360, the steam cylinder is placed at the bottom between the housings, which are strongly ribbed castings, extending from the bed plate, upon which they rest, up to the lower head of the air cylinder, and containing the cross head guides.

Where increased efficiency in the use of steam is desired, compound engines are used; about the only objection that could be raised to their use is the more serious results that would follow the shutting down of two air tubs in case of accident.

Some builders have overcome this objection by building the high pressure and low pressure engines separate, as shown in fig. 1,360, the only connection being a receiver between the two steam cylinders. The high pressure engine is controlled by a governor with variable speed device.

It has been found that the low pressure engine, when running compound, requires no governor to insure a uniform speed of both high and low pressure engines. Some provision is made on the low pressure side, when running as an independent engine, to control the steam admission as by governor, or by hand adjustment.

In order that the variable pressure due to expansion of the steam and compression of the air may be approximately balanced, blowing engines are sometimes constructed having a phase difference of ninety degrees in the sequence of strokes at the steam and air ends. This type is known as the *quarter crank engine*.

About the only advantage of this arrangement is that a somewhat smaller fly wheel will suffice than where the steam and air cylinders are placed tandem, as here the maximum power exerted in the steam cylinder before cut off is not balanced by any resistance in the air cylinder, and the surplus energy is stored in the fly wheel to be given out again when the resistance in the air cylinder has increased beyond the power exerted in the steam cylinder.

The quarter crank engine is objectionable on account of: 1, higher first cost; 2, greater size; 3, multiplicity of parts, and 4, having to transmit the power for operating the air piston through cross heads, connecting rods and shaft resulting in a decrease of mechanical efficiency, extra wear on bearings.

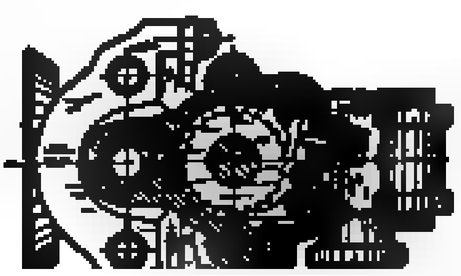
The balancing of the air and steam pressures due to the quarter crank arrangement, while very nice in theory is of no practical value beyond serving as a "talking point" for salesmen.

CHAPTER 24

ROCK DRILLS

The rapid tendency toward more rapid and more economical methods of operation makes itself felt, especially in the mining field in the demand for high speed, high duty rock drills with large capacity.

FIG. 1,362.—Wickes rock drill for either steam or air. The tripod legs are made with a telescopic extension which allows of an adjustment on uneven ground and by loosening a single bolt, either of the front legs may be made to swing into any desired position.



The art of drilling rock has reached such a state of efficiency that the manager of a project involving different kinds of work, usually employs a different type of drill for each kind.

Drills may be operated by either steam or compressed air, the latter being found convenient especially in confined places such as mines. The mechanism is arranged so as to deliver two motions to the drill.

1. Reciprocating;
2. Rotating.

Air Consumption of "Chicago Slogger" Drills at Sea Level

Size of drill	2¾ inches —C 2	3 inches —D 2	3¼ inches —B 2	3¾ inches —F 2
Air pressure pounds per square inch	Air consumption at sea level of one drill in cubic feet of free air per minute			
60	80	90	100	115
70	92	104	114	130
80	102	114	130	141
90	112	128	142	160
100	124	140	156	176

Figs. 1,363 to 1,365.—Sectional views of Wickes rock drill showing construction of ratchet, valve and feed screw. A slide valve is used with tappet motion, which depends on the movement of the piston.

FIG. 1,368.—Sectional view of Leyner-Ingersoll drill showing mechanism. *At the top* (left hand side) is seen the butterfly valve, and directly below the ratchet, and to the right, the cylinder anvil, air and water supply nozzle, and hollow drill. *At the bottom* is seen the feed screw, crank and feed nut.

FIGS. 1,371 and 1,372.—"Chicago slugger" drill valve motion. The various paths followed by the air are shown by the arrows. In fig. 1,371 auxiliary valve is in the position in which it was thrown by a shoulder on the drill piston as it approached the end of its backward stroke. It will be noted that live air passes through the hole A, in the auxiliary valve, to one end of the main valve, the other end of the main valve being placed in communication with the exhaust by the recess in the valve, the auxiliary valve resembling an ordinary D slide valve. With the auxiliary valve in the position, as shown, the main valve was thrown and held positively in position, to admit live air to the rear end of the drill cylinder and allow the air to escape from the front end. Now when the drill has nearly completed its forward stroke and is about to strike its blow, another shoulder on the drill piston strikes the auxiliary valve and throws it into the position shown in fig. 1,372. In this position it connects the front end of the main valve with the exhaust, but there is no opening at the opposite end of the auxiliary valve, corresponding to port A, to admit live air to the rear end of the main valve. The auxiliary valve merely closes the end of the passage which was previously in communication with the exhaust. There is a small vent hole B, which opens from the live air supply. With the auxiliary valve closing the exhaust end of the passage, this small supply of air builds up pressure enough to throw the main valve, a slight interval of time being required to do it. This slight delay in the throwing of the main valve insures full pressure upon the rear end of the piston, and a free exhaust from the front end, until it has had ample time to deliver its blow with full force upon the rock in which it is drilling.

FIGS. 1,367 to 1,370.—*Text continued.*

rotation or oscillation of the wings with the trunnion as a center, and is actuated by the unbalancing of pressure on each wing alternately. The supply and exhaust ports open into the faces of the slot, and the wings of the valve close the ports by seating flat over them against the sides of the slot. There is a separate supply and exhaust port and passage to each end of the cylinder, making two supply and two exhausts in all. The two supply ports open directly opposite one another at one end of the valve chest slot, with the two exhaust ports similarly placed at the other end. The valve trunnion in place closes any opening direct from supply to exhaust through the slot. Adjustments are such that when one face of one wing of the valve rests over and closes the supply port to one end of the cylinder, the opposite face of the opposite wing closes the exhaust port from the opposite end of the cylinder. This leaves open the supply port to one end of the cylinder and the exhaust port from the other end—the condition for one-half of the operating cycle.

NOTE.—***Air consumption of rock drills.*** (*Ingersoll-Rand Co.*) The following table gives the free air consumption per minute of rock drills of various cylinder diameter for fair conditions in rock of ordinary hardness:

Gauge pressure, lbs. per sq. in.	CYLINDER DIAMETER OF DRILL, INCHES											
	2	2¼	2½	2¾	3	3¼	3½	3⅝	3⅞	4¼	5	5½
60	50	60	68	82	90	96	100	108	113	130	150	164
70	56	68	77	93	102	108	113	124	129	147	170	181
80	63	76	86	104	114	120	127	138	143	164	190	207
90	70	84	95	115	126	133	141	153	160	182	210	230
100	77	92	104	126	138	146	154	168	174	199	230	253

Thus, in operation the drill strikes a series of blows and rotates at the same time.

The mechanism consists essentially of a power piston working in a cylinder, having at one end a ratchet device and at the other, a rod extending through a stuffing box, and having at its end a chuck to receive the drill. A valve motion, similar to the direct connected pump types, is provided to secure the reciprocating motion of the piston and drill. The cylinder is attached to a feed device, and the latter mounted on a weighted tripod.

Fig. 1,373.—Leyner-Ingersoll drill set up on column, showing arrangement when water tank is used. Water under pressure is required and the pressure may be anything over twenty pounds and not higher than the air pressure. The water can be brought to the drill in any manner that is most convenient. Some run a small water pipe along with the air pipe, obtaining the supply either from outside the mine, or from the water in the mine, arrangements being made on the different levels so as to provide suitable average pressures. Others have one or more central tanks, each supplying several drills. In tunnel headings a suitable tank is often run in on a car. Where none of these methods is available a small individual tank for each drill is used. The tank usually employed weighs 70 pounds, is 3 feet 3 inches in height and 12 inches in diameter, and holds about 17 gallons. A smaller tank of about 6 gallons capacity can also be furnished, if preferred. Compressed air is admitted to the tank, and this forces the water to the drills under the same pressure as the air used to operate the drills, if taken from the same air line. The air which is admitted to the water tank is not the air which combines with the water and cleans the drill holes. This air comes from the drill. The water is carried from the water pipe line, or from the water tank, through a small water hose about 50 feet in length, then through a water regulating valve and a short piece of hose to the backhead of the drill. The water passes through the backhead plug and water tube, into the shank of the drill steel, where air from the drill mingles with it, and both pass on through the drill steel to the bottom of the drill hole. A swiveled connection is attached to the backhead, to which the short length of water hose is attached. The air hose is attached to the swivel connection at the valve chest. This method permits the hose to lie in a natural position close to the drill, and eliminates the necessity for a bend in the hose itself.

Fig.
cip
Th
va:

Cylinder	$2\frac{1}{4} \times 3\frac{1}{4}$	$2\frac{3}{4} \times 5\frac{1}{4}$	$3\frac{1}{4} \times 5\frac{3}{4}$	$3\frac{1}{4} \times 5\frac{3}{4}$
Length of feed	20	26	25	26
Depth of hole drilled, feet	8	14	18	20
Number pieces drill steel in set	6	7	9	10
Size boiler for steam, *h.p.*	6	8	10	10

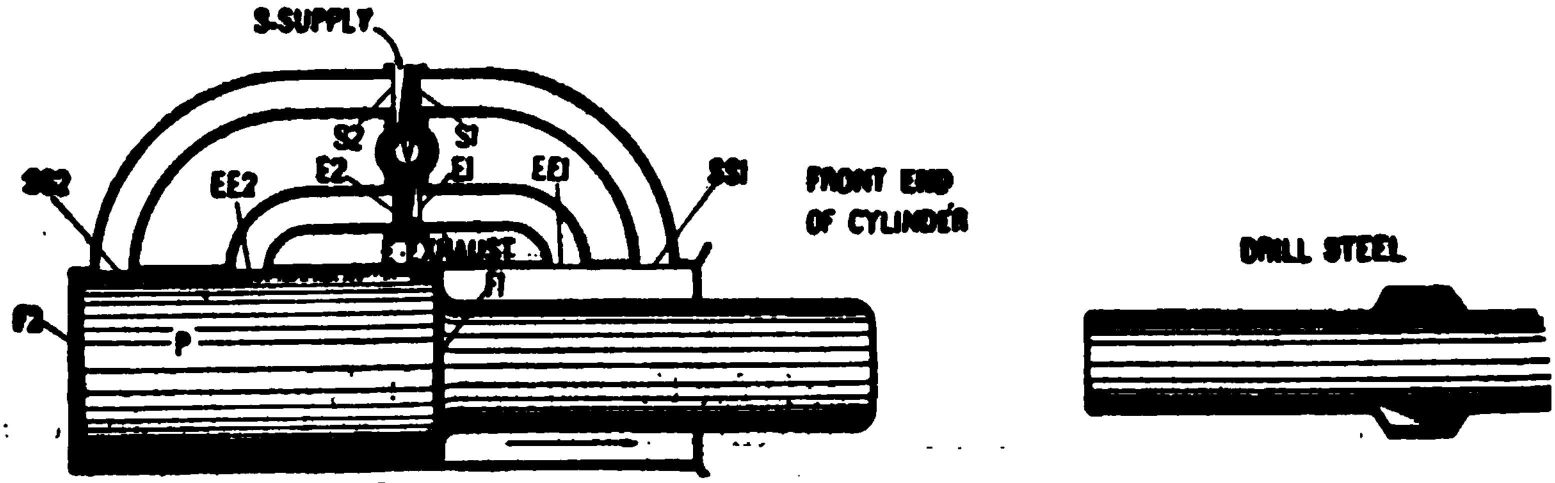
Fig. 1,362 shows the general appearance of a typical rock drill embodying the features just described. Figs. 1,363 to 1,365 are sectional views of the same drill showing construction.

In the Leyner drill air and water combined are passed through hollow drill steel.

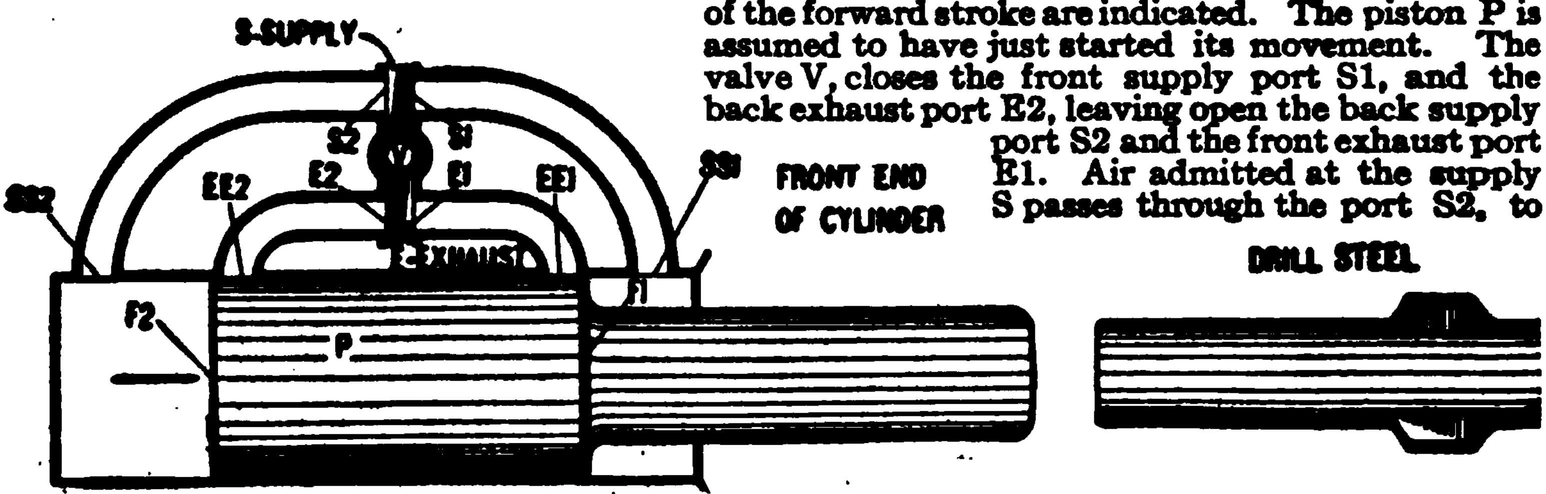
The application of water and air is for the purpose of: 1, laying the dust; 2, cleaning the holes of the rock cuttings while drilling, and 3, keeping the drill bit cool.

FIGS. 1,375 to 1,377.—Ratchet parts of Leyner-Ingersoll water drill. The ratchet is a separate ring held in place between the rotation washer and the back head, and kept from turning by a large pin. It is thus quickly removed, should occasion require, by taking off the back head.

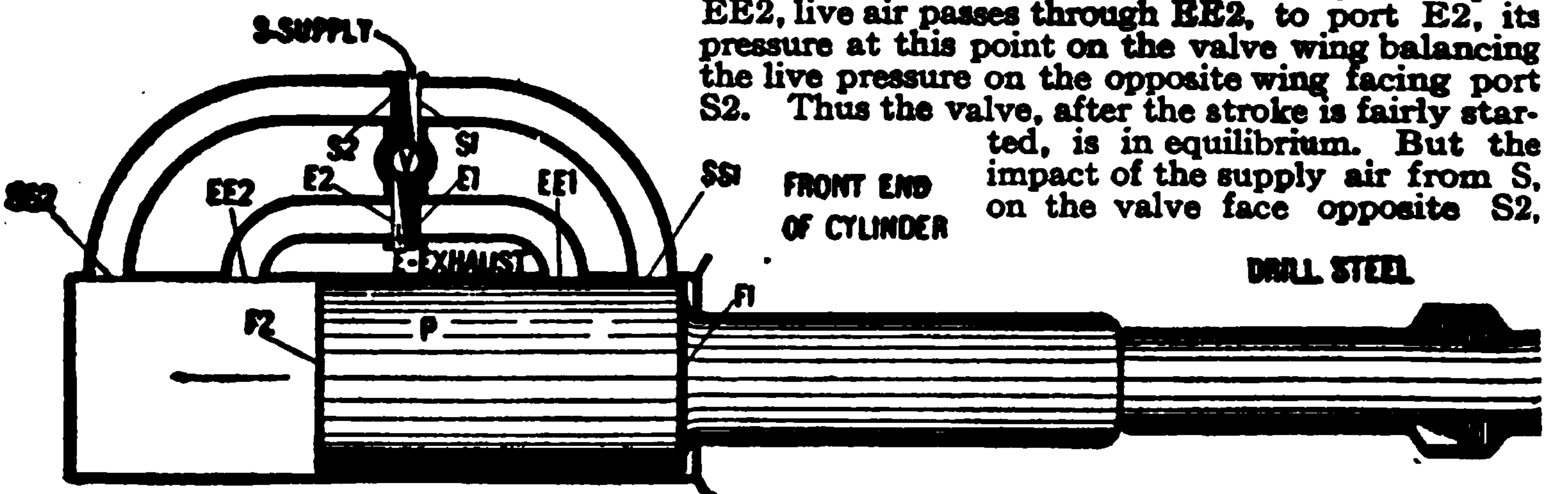
FIGS. 1,378 to 1,380.—Ingersoll-Rand butterfly drill slip "rotation." *In construction*, the head of the rifle bar, carrying the pawls, plunges, and springs, fits inside the ratchet ring. The latter is held by friction between the rotation washer and the back head, pressure being exerted by the cushion springs. By varying the spring tension, the friction effect can be adjusted to various conditions. Broken or twisted rifle bars are impossible with this rotation, even in the worst ground.



FIGS. 1,390 to 1,394.—Diagrams of Leyner-Ingersoll drill butterfly valve showing valve positions and pressure conditions at the three critical points in one stroke. The forward stroke is here illustrated, but the cycle of operations is the same for the back stroke, so no separate explanation is made for the latter. In figs. 1,390 and 1,391, the conditions at the beginning of the forward stroke are indicated. The piston P is assumed to have just started its movement. The valve V, closes the front supply port S1, and the back exhaust port E2, leaving open the back supply port S2 and the front exhaust port E1. Air admitted at the supply S passes through the port S2, to



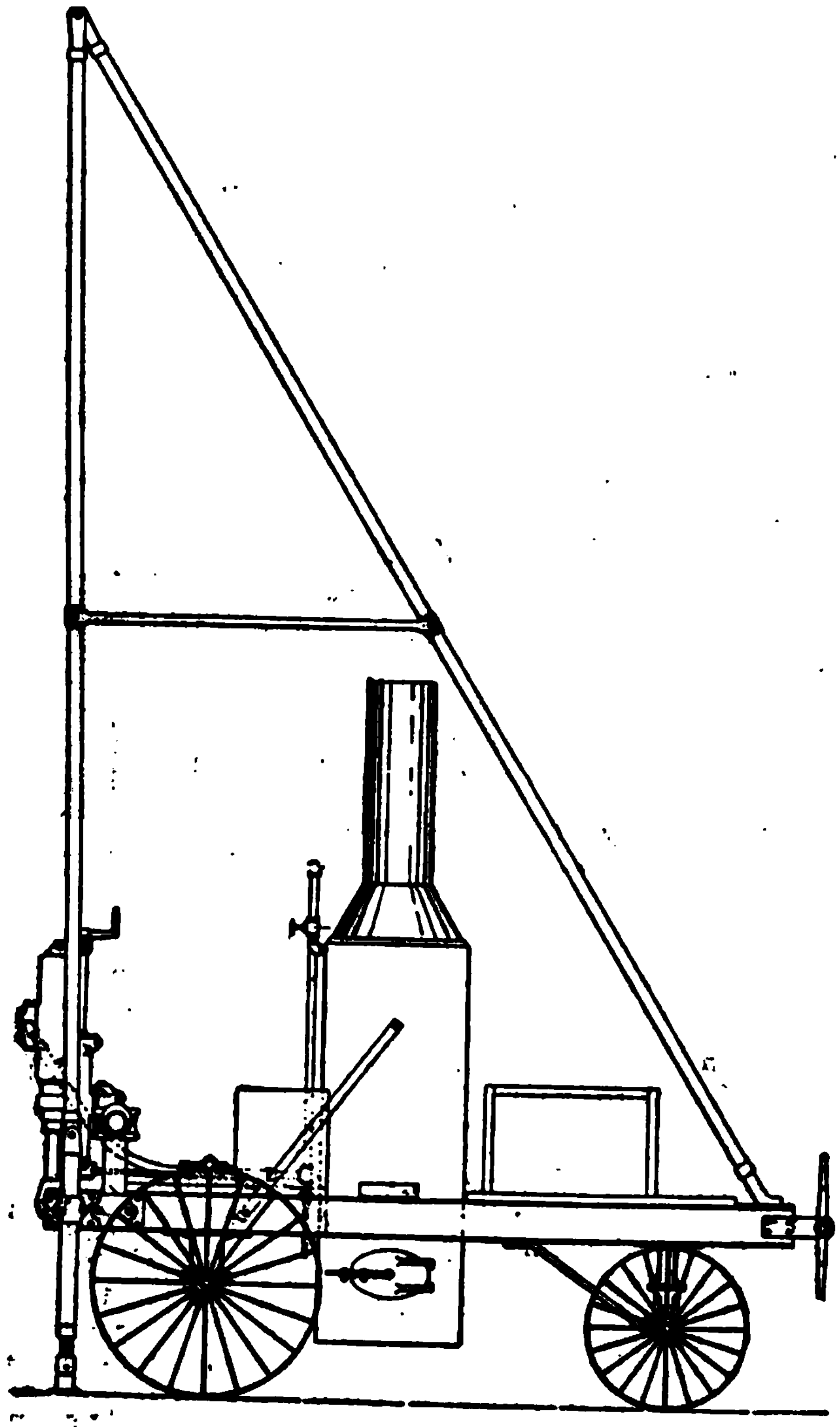
the cylinder port SS2, forcing the piston forward, while the exhaust passes out through the cylinder port EE1, and the port E1. In this particular position of the piston the cylinder port EE2, is covered by the piston, and the valve V is in an unbalanced condition, live air holding it to its seat over port S1. However, the moment piston P, uncovers cylinder port EE2, live air passes through EE2, to port E2, its pressure at this point on the valve wing balancing the live pressure on the opposite wing facing port S2. Thus the valve, after the stroke is fairly started, is in equilibrium. But the impact of the supply air from S, on the valve face opposite S2,

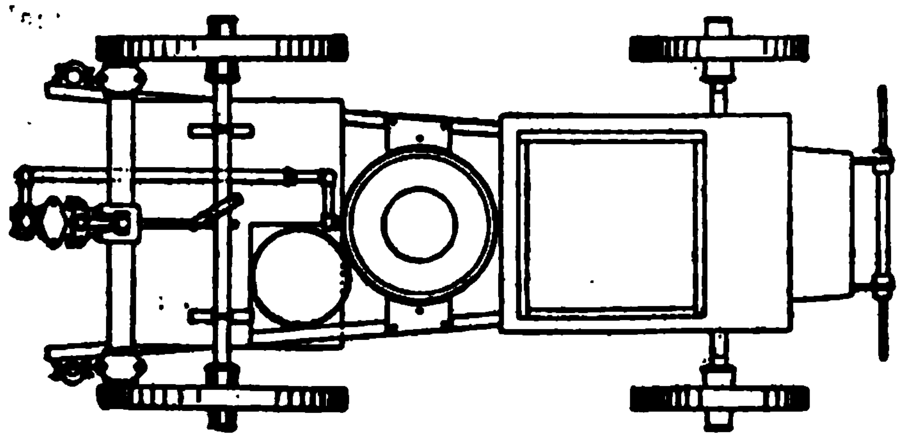


holds the valve stationary, with ports S2 and E1, fully open. Now in figs. 1,392 and 1,393 the conditions are shown near the end of the forward stroke. The valve V, is in the same position as in fig. 1,390. The face F1, of the piston P, has passed the cylinder port EE1, just closing it and shutting off the exhaust from this end of the cylinder. Live pressure, however, is still exerted through S2 and SS2, against the piston face F2, forcing P forward and causing cushion pressure in the clearance between piston face F1, and cylinder head. This cushion pressure, communicated through cylinder ports SS1, to port S1, and exerted at S1, on the balanced valve, throws the valve V, to the position shown in fig. 1,394. Here the back stroke is just beginning, and live air enters through S1 and SS1, with exhaust escaping through EE2 and E2, to E. The valve V, is unbalanced, but will become balanced the moment piston face F1, passes cylinder port EE1, which will admit live air through EE1 to E1, on the valve.

HAMMER CYLINDER
or BARREL

FIGS. 1,395 and 1,396.—Denver, Waugh type, stoping drill; sectional view showing operation. The positions in figs. 1,395 and 1,396 differ by 45°. ***In operation,*** when the hammer reaches the end of its rearward stroke, as in fig. 1,395, the exhaust ports, 5, in the front of the hammer cylinder, have been cleared by the piston head and the pressure there relieved through horizontal exhaust ports, 4. At this point the neck or thin part of the stem registers with inlet ports, 2, which are connected with horizontal ports, 1, and admit live air against the rear of the piston head, driving it forward. As the stem of the piston hammer is always in contact with live air, the pressure is on the total area of the hammer. When the piston on the forward stroke reaches the point where the shoulder of the stem covers the ports, 2 in the bushing, live air is shut off from the rear of the piston head and the piston hammer continues the stroke on expansion plus the pressure on the end of the stem. When the piston hammer uncovers the exhaust ports, 3, fig. 1,396, the air back of the head is exhausted to the atmosphere and the stroke is completed by the momentum of the hammer and the pressure on the stem. The piston hammer thus runs on expansion for half the stroke, and as it exhausts at a much lower pressure, comparatively little air is used. As the piston hammer approaches the tappet on the end of the forward stroke, fig. 1,396, its stem clears inlet ports, 6, admitting live air to the front of the cylinder through horizontal inlet ports, 7, and vertical forward ports, 8. Ports 6 remain open until the piston has delivered its blow and returned about 1/7 of its stroke, when the stem of the piston again covers ports 6, which shuts off the live air from the front of the hammer cylinder and the piston continues on its rearward stroke expansively. When the front of the head again uncovers vertical ports 5, this air is allowed to escape to the atmosphere and the momentum of the hammer carries it to the end of the rearward stroke which brings the head within about one-half inch of the bushing, where it cushions on the incoming live air that is again entering ports 1 and 2. From this, it is evident that all of the live air that is used in returning the piston is that which is admitted by the stem of the piston at ports No. 6 just previous to and after the blow has been delivered, which is equal to about 1/4 of the full stroke. Part of this air is used to fill the long horizontal ports 7, and is therefore not actually used until expansion takes place. The difference between the distance the piston travels rearwardly before this live air is cut off, or about 1/7 of the stroke, and the distance it travels after the air is cut off, until port 5 is uncovered, determines the amount of expansion, which is nearly 1/4 of the full stroke, consequently this air is exhausted at a greatly reduced pressure.

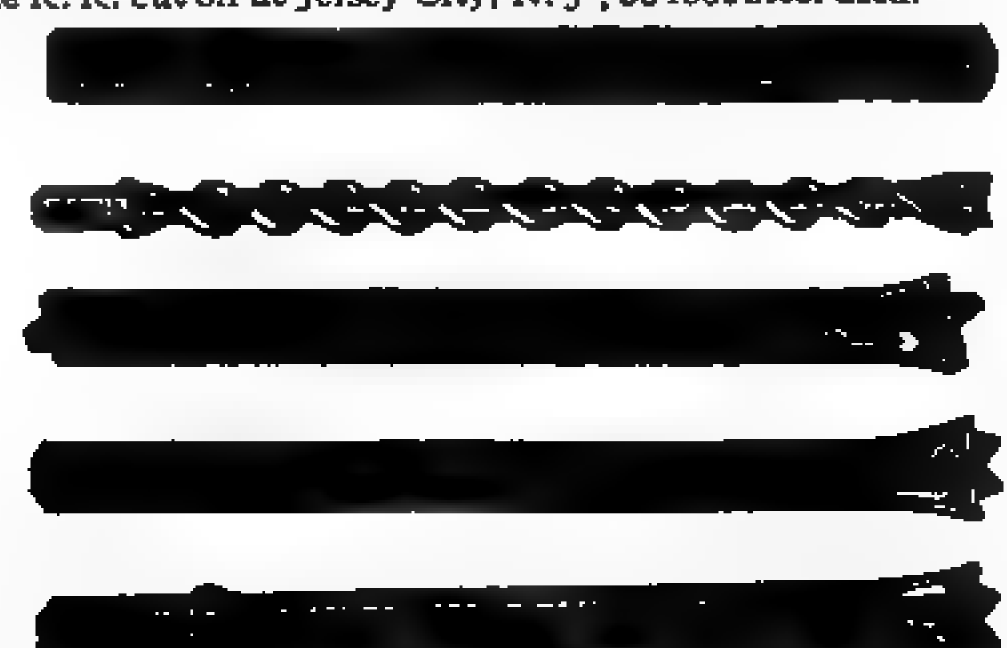




FIGS. 1,397 and 1,398.—Portable deep hole drill wagon mounting for a single drill. The mounting consists of steel wheels and axles supporting a structural iron frame on which the boiler and the drill mounting are located. The wheel gauge is about 6 feet, and the wagon is provided with tongue, whiffle trees and yoke for a two horse team. For making short moves such as from hole to hole, a small winch is provided in front, on which a rope can be wound. This rope passes over a sheave, which is fastened to a "dead-man," the other end being fastened to the end of the tongue. One or two men can easily handle the wagon over very rough ground. The drill mounting consists of a horizontal bar fastened in brackets on the structural iron work. On this bar the drill is mounted in the regular way by means of a Sergeant clamp, which gives it universal adjustment so that the holes can be pitched in any desired angle. The frame extends back beyond the center of the drill, and is provided with two jack screw supports which are set up for supporting the end of the wagon during the actual drilling. For convenience in handling the steels, for deep holes, the wagon is provided with a 24 foot, three legged, heavy pipe derrick, so that the steel can be lifted in and out of the hole, by means of rope and tackle. It will handle steels 30 to 40 feet long. The boiler is of the vertical type, with damper regulator on the smoke stack and live steam forced draft. Under the wagon, and convenient to the firing place, is located the coal bin. An iron water tank is also provided and is connected to the injector pipe on the boiler. Connections from drill to boiler are short, so that there is very little loss in transmission. This makes an inexpensive yet efficient mounting for a large variety of work, especially quarrying and moderate railroad cuts. The equipment is easily handled by two men and is capable of doing considerably more work than the same two men could accomplish with a tripod drill, especially in places where the holes are spaced far apart. Two men easily handle a 4¼ inch drill, which they could not do otherwise except under extremely favorable conditions. When furnished for air power operation this machine is equipped with suitable air reheater in place of the boiler outfit.

Another feature of the Leyner drill is the principle of the hand hammer blow mechanically applied. The drill steel does not reciprocate, but rests loosely in the chuck, and is struck by the hammer. The cutting end of the steel is close to or against the rock at all times, hence the weight moved, that of the hammer

FIG. 1,399.—Ingersoll-Rand-6 foot feed drill wagon at work drilling holes for cement grouting bridge over Erie R. R. cut off at Jersey City, N. J ; 50 foot steel used.



FIGS. 1,400 to 1,404—Various rock drill bits. Fig. 1,400, hexagon hollow, shanked; fig. 1,401, hexagon hollow, shankless; fig. 1,402, cruciform shankless, fig. 1,403, solid hoist with shank; fig. 1,401, hollow steel for making drill bits.

alone, is light and constant at all times regardless of the length of the drill steel.

Fig. 1373 shows the Leyner drill in position to operate with portable water system and fig. 1366 a sectional view of the mechanism. Other details of this drill are shown in figs. 1375 to 1377.

For a large variety of deep hole work it has proved convenient as well as economical to have a drill and boiler mounted together on a truck, the arrangement being called a portable wagon mounting for a single drill, as shown in figs. 1,397 and 1,398. The mounting here shown is suitable for a 3½ or 4½ rock drill with two to four feet feed.

FIG. 1,405.—Ingersoll-Rand 15 foot feed drill wagon in use on the P. & L. E. R. R., drilling 15 foot holes, 4 to 5 inches in diameter at Youngstown, Ohio. The running gear is of the three point suspension type, comprising heavy steel wheels with flanges for traveling on a 6 foot 3 inch gauge track and provided with power traction, through the medium of a chain drive from a reversible steam engine, mounted on the wagon. The engine is about 7 horse power and, while not intended for continuous traction, supplies sufficient power for moving the wagon from hole to hole. A hand wheel and steering gear are also provided at the front end of the wagon. The drilling engine and its guides, derrick for hoisting steel, etc., are carried on the turn table, which is mounted just past the center of the wagon, at the rear end. This turn table runs on balls. A system of sheaves and ropes suspends the drill from the top of the derrick, from where the ropes lead to a reversible hoisting engine which is fastened to the back of the drill guides. Powerful hand brakes on the hoist regulate the feeding of the drill. Heavy jack screws are provided for supporting and steadying the drill frame during drilling operations. The swinging of the turn table is accomplished by means of a geared hand winch and rope. A half circle, 10 feet in diameter, is described when swinging, and the holes can be placed anywhere on this line. The drilling machine furnished with this rig is either a 5¼ inch submarine drill, or a 4¼ inch submarine drill, and will drill a hole from 3½ to 5 inches in diameter. A steam pump for feeding water to the drill by means of a pipe suspended from the top of the derrick is a part of the equipment.

CHAPTER 25

SEMI-PORTABLE AND PORTABLE ENGINES

There are some kinds of work that require a good steam power, and yet the character of the work is such that the outfit is not

FIG. 1,406.—Leffel semi-portable overmounted engine with locomotive boiler.

moved about sufficiently to warrant the investment in a traction engine. In such cases there is a saving in cost by the use of a plain engine and boiler mounted on wheels, or on skids, forming a self contained unit which can be easily moved.

FIG. 1,407.—Leffel semi-portable undermounted engine with locomotive boiler.

FIG. 1,408.—Leffel semi-portable overmounted engine, with Clyde boiler.

FIG. 1,409.—Leffel portable engine with vertical boiler.

FIG. 1,410.—Farquhar portable engine and boiler belted to portable saw-mill.

Where skids are used the engine is called *semi-portable*, and in the case of wheels, *portable*.

For service requiring the engine to be moved only at long intervals as in saw mill work, etc., the semi-portable type is frequently used, as shown in fig. 1,406. With either type the usual style of boiler used is the locomotive, as in fig. 1,406, or a return tubular with internal cylindrical fire box as in fig. 1,408. The latter has a fire box extending the length of the boiler, and is designed especially for burning wood, etc. The boiler shown in the cut has no grate bars but instead short draft tubes running from the fire box through the bottom of the boiler.

With machines of this class ordinary slide valve engines are used, as the conditions of service are not such as would call for special economy in the use of fuel. In many cases a feed pump is attached to the cross head

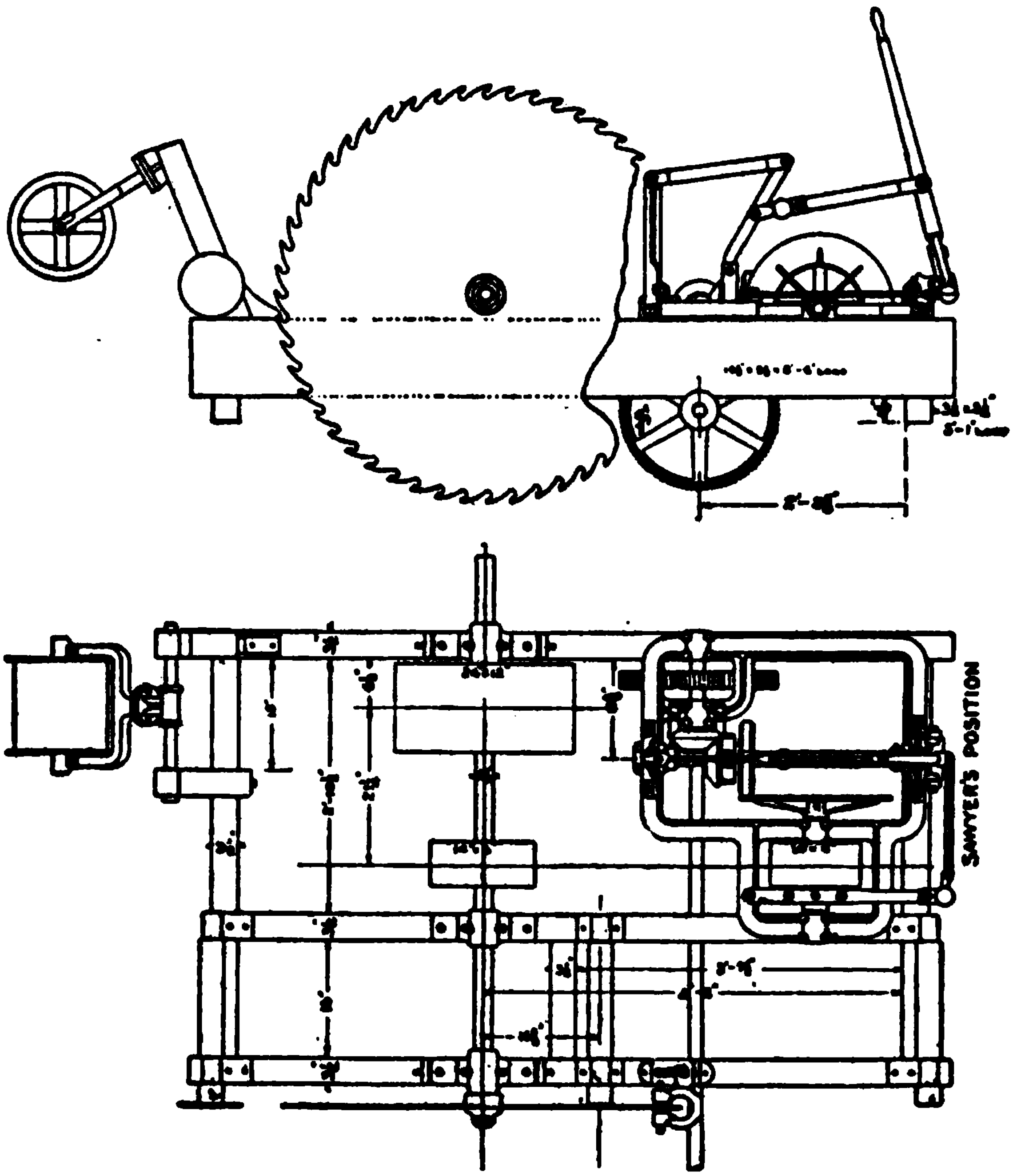
FIG. 1,411.—Semi-portable undermounted engine and boiler with saw mill, edging saw, slab saw, lumber hack, arbor extension and cable feed.

FIG. 1,412.—Semi-portable undermounted engine and boiler with saw mill, edging saw, cut off saw, rollers, line shaft, arbor extension, and cable feed.

of the engine, the water passing through a heater before entering the boiler.

There should be in addition, an injector, as an auxiliary, the pump being used for regular feed, on account of the economy in heating the water by the exhaust steam, and because a constant feed is better than one intermittent, as with the injector.

A well proportioned pump, feeding through a heater, forms the ideal method of supplying a boiler, as the water is delivered at approximately



FIGS. 1,413 and 1,414.—Russell saw mill friction feed. *In operation*, the friction wheel moves horizontally across the disc, reversing the motion as it passes the center. Thus the forward motion as well as the gigback is accomplished by the use of but one friction wheel, and the variation in speed for either motion is from nothing to 9 inches per revolution of saw. This entire range requires but a slight movement of the lever. The sawyer is thus enabled to feed the log into the saw up to the full capacity of the engine power, and to slow up when he comes to a knotty place in the log, increasing the speed again when this place is passed. The friction disc being on a separate shaft from that of the saw arbor, and simply driven from a pulley on it, no thrust is transmitted to the saw by forcing the friction wheel against the disc.

the same rate it is used, thus avoiding undue fluctuations of steam pressure, and strains due to too rapid cooling.*

FIG. 1,415.—Leffel portable vertical engine and boiler; view showing engine and boiler turned on side ready for moving. Mounted arrangement consists of two iron saddles or brackets with the steel axles made part of same, securely attached near mid length of Boiler. When at work the engine is set upright on base, and the tongue and wheels removed, which is easily and readily done. Built in sizes 3 to 7 horse power cylinder 3¼×6 to 5×8; 240 to 220 R. P. M. respectively.

*NOTE.—The cost of production in the manufacture of lumber is at the present time an important item, yet it is still customary in many small and medium mill plants to follow the old and expensive method of edging boards and planks on the mill carriage, consuming considerable of the entire crews' time each day in doing what one man can do alone on a single edger without interfering with the operation of the mill, together with the saving in cost of labor resulting from the use of a single saw edger. Some single edgers are built in three sections to facilitate moving. In one edger of this type, taken to illustrate the proportions and construction, the saw mandrel is 1¾ inches in diameter and 6×6 inch drive pulley. It is furnished with 32 feet of track, which accommodates a carriage 14 feet in length by 18 inches in width. The carriage has four sets of trucks consisting of 3¼ inches diameter flange wheels on steel axles running in babbitted self-oiling bracket bearings. This edger is built either right or left handed and takes a 14 inch saw which should run at a speed of 2,600 revolutions per minute.

*NOTE.—Under no circumstances should cold water be fed into a boiler, yet there are on the market outfits in which no provision is made for heating the feed water. The accompanying illustrations show the usual construction of semi-portable, and portable engines. The wheels of the latter are of iron substantially made and have wide rims to distribute the considerable weight carried, and prevent them sinking in soft roads. A portable engine is provided with a tongue, so that it can be drawn by horses when it is to be moved. Powerful brakes are fitted for control in descending grades. Portable engines are extensively used in very hilly countries where threshing jobs are small, or when an inexpensive outfit is desired.

CHAPTER 26

LOCOMOBILES

If the energy latent in fuel could be completely transformed into work, a horse power hour could be produced on less than one-fifth pound of coal.

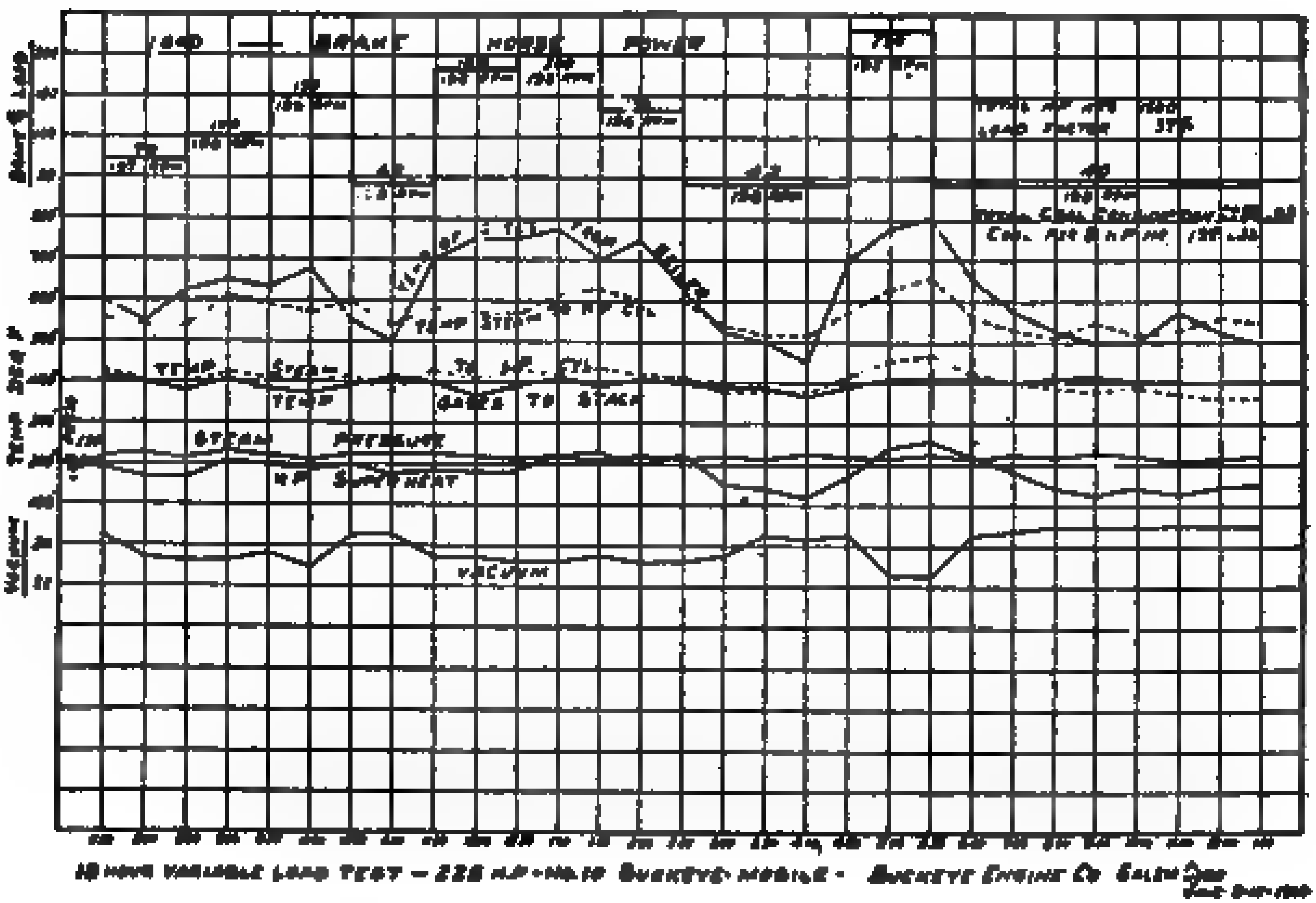
From the beginning of the history of the steam engine, the best engineering talent has devoted itself to improving the efficiency of steam power plants, that is, increasing the amount of power that can be obtained from a given quantity of fuel.

James Watt invented the separate condenser whose only value was to make a pound of coal go farther in generating power. Since his day pressures have increased from atmospheric pressure to 250 lbs. for general use and in special cases up to 500 or more lbs.

More than a century of engineering research and experience has demonstrated that proper furnace and boiler design, insuring complete combustion of the fuel and absorption of the heat by the boiler multiple expansion of high pressure steam; reduction of cylinder condensation through the use of steam in a superheated condition; elimination of radiation loss by the effective jacketing of all parts containing live steam; recovery of waste heat through a feed water heater and the reduction of back pressure by means of a condenser are all effective means of reducing waste of heat to the lowest figures.

It was the clear recognition and intelligent application of these principles by able foreign engineers that gave to the world the special form of high efficiency combined engine and boiler plant known as the "locomobile." This invention, because of its remarkable economy, ease of attendance and reliability as well as its adaptation to all situations where power is needed, has met with wide spread application in Europe and its dependencies. In fact the locomobile seems to be well known to nearly every part of the world except the United States. (One German firm alone has built nearly 1,000,000 h. p. of these machines, in sizes up to 800 h. p. in a single unit.) For the locomobile is in no sense an experiment but rather the evolution of the conventional power plant into its natural and simplest form, a carefully chosen combination of tried and demonstrated principles.

FIG. 1,416.—Longitudinal section of Buckeye-mobile showing boiler, superheater, engine, reheater, and general arrangement of parts. It develops a horse power hour on from 1 to 2 lbs. of coal (depending upon the quality).



FIGS. 1,417 and 1,418.—Performance curves of Buckeye-mobile. Fig. 1,417 represents a 24 hour test of a 225 horse power unit in which the load varied from 70 horse power up to 200, then down to 40, up to 230 and down to 40 for the last seven hours of the test period. The average load was only 37 per cent of the rated capacity of the unit but notwithstanding this low load factor the machine consumed less than two pounds of coal per brake horse power hour.

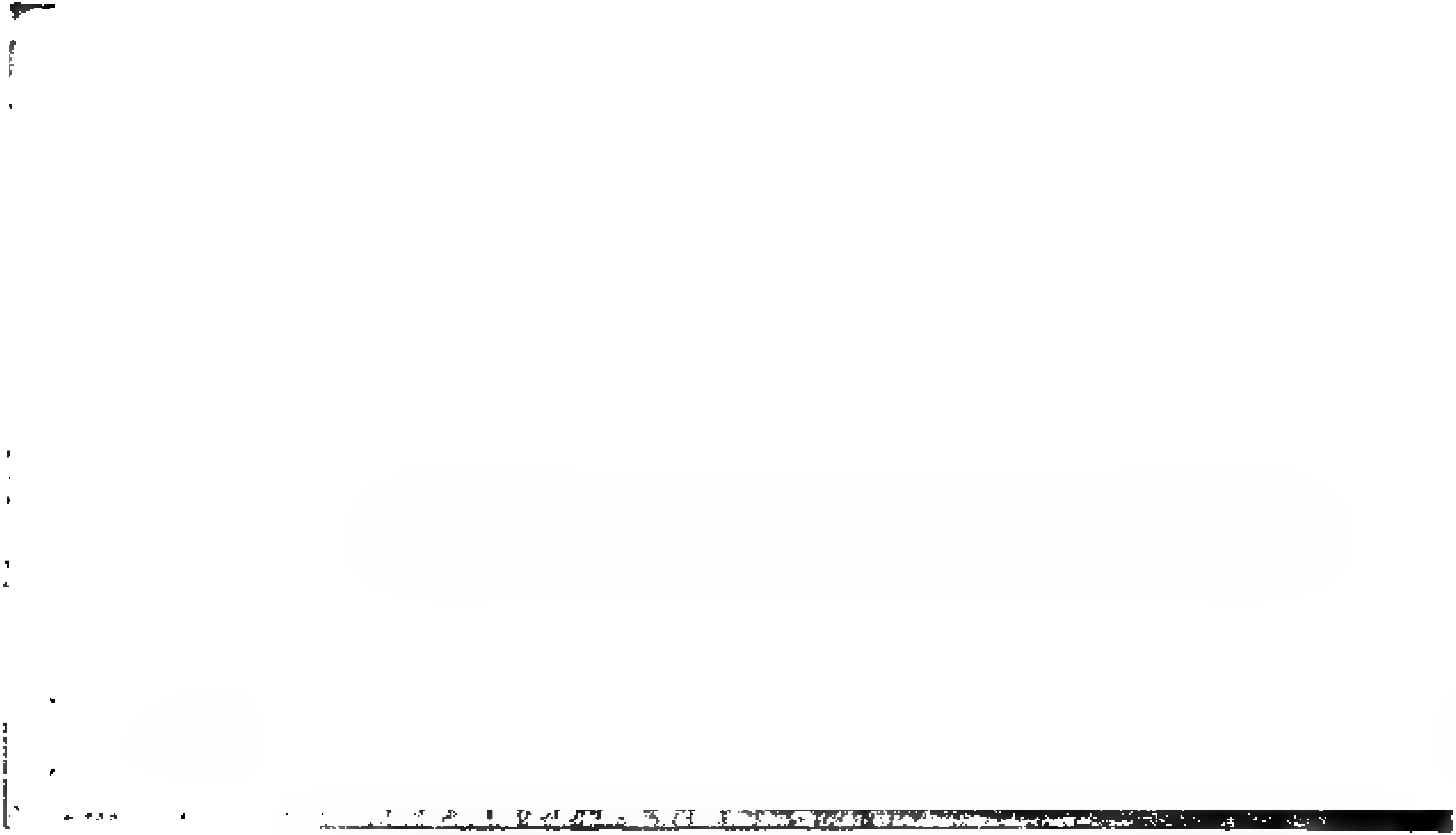
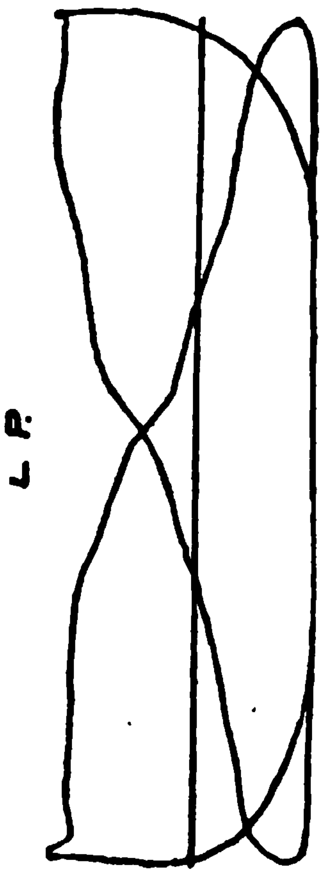
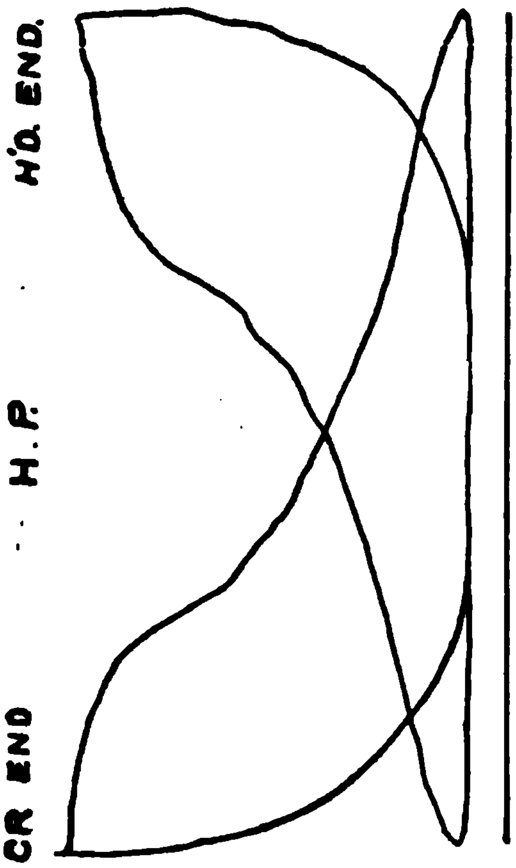


FIG. 1,419.—View of Buckeye-mobile with engine dismantled, smoke box and jacket removed showing construction and general arrangement of superheater, reheater and connections between engine and the two heaters.

engine economy.





FIGS. 1,420 and 1,421.—Indicator diagrams for Buckeye-mobile. Fig. 1,420 *h.p.* diagrams; fig. 1,421, *l.p.* diagrams.

Accordingly, a locomobile may be defined as *a self contained engine and boiler with all the auxiliary apparatus, necessary to secure maximum economy.*

The accompanying illustrations show a locomobile representing American practice, being called the Buckeye-mobile because it is made by the Buckeye Engine Co.

It consists of a self-contained power plant designed to use high pressure superheated steam. In its elements it comprises: 1, an internally fired tubular boiler of the non-return or "gun-boat" type on which is mounted, 2, a compound engine; 3, a well insulated sheet metal smoke box encloses, 4, a tubular super-heater, both engine cylinders, 5, all steam piping and valves, and 6, a secondary superheater which imparts heat to the steam as it passes from the high to the low pressure cylinder. The engine exhausts through a closed feed water heater into a jet condenser which is provided with a rotary air pump. This air pump and the boiler feed pump are so located as to be readily belt driven in a most

economical manner from the engine shaft. The general assembly is shown in sectional view in fig. 1,416.

Boiler.—The boiler is of the internal corrugated furnace fire tube type. The furnace tubes and tube sheets constitute the entire active heating surface, therefore deterioration and need of repairs are confined to these parts which are attached to the boiler shell by studs and nuts, making them readily removable for inspection and cleaning. Sediment collects in the bottom of the boiler (the coolest part) where it is not liable to bake. The boiler is constructed for 225 lbs. steam as shown in fig. 1,416.

Superheater.—This consists of a single coil of seamless steel tubing as illustrated in fig. 1,416. The steam passes through it in a direction

FIG. 1,423.—Sectional view of Buckeye-mobile showing temperatures at various points of the system, viz.: gases in furnace 2,500°; gases to superheater 800°; gases to smoke stack 425°, steam to superheater 396°, steam to *h. p.* cylinder 625°, exhaust from *h. p.* cylinder 325°, steam to *l. p.* cylinder 430°, exhaust from *l. p.* cylinder 140°, feed water to heater 70°, feed water to boiler 125°, injection water 70°, hot well 105°.

counter to that of the hot gases. The flexibility of the coil prevents any expansion strains in the high pressure steam valves or fittings.

Engine.—The engine is of the compound center crank piston valve type of extremely simple and rugged construction. The bed plate is rigidly bolted at its main bearing end to a massive saddle which spans about one-third of the boiler's circumference. The guide barrel end rests on a smaller saddle and is free to slide thereon, thus relieving the bed plate from the effects of boiler expansion.

Reheater.—The steam on its way from the high pressure to the low pressure cylinder passes through a reheater built up of a large number of small

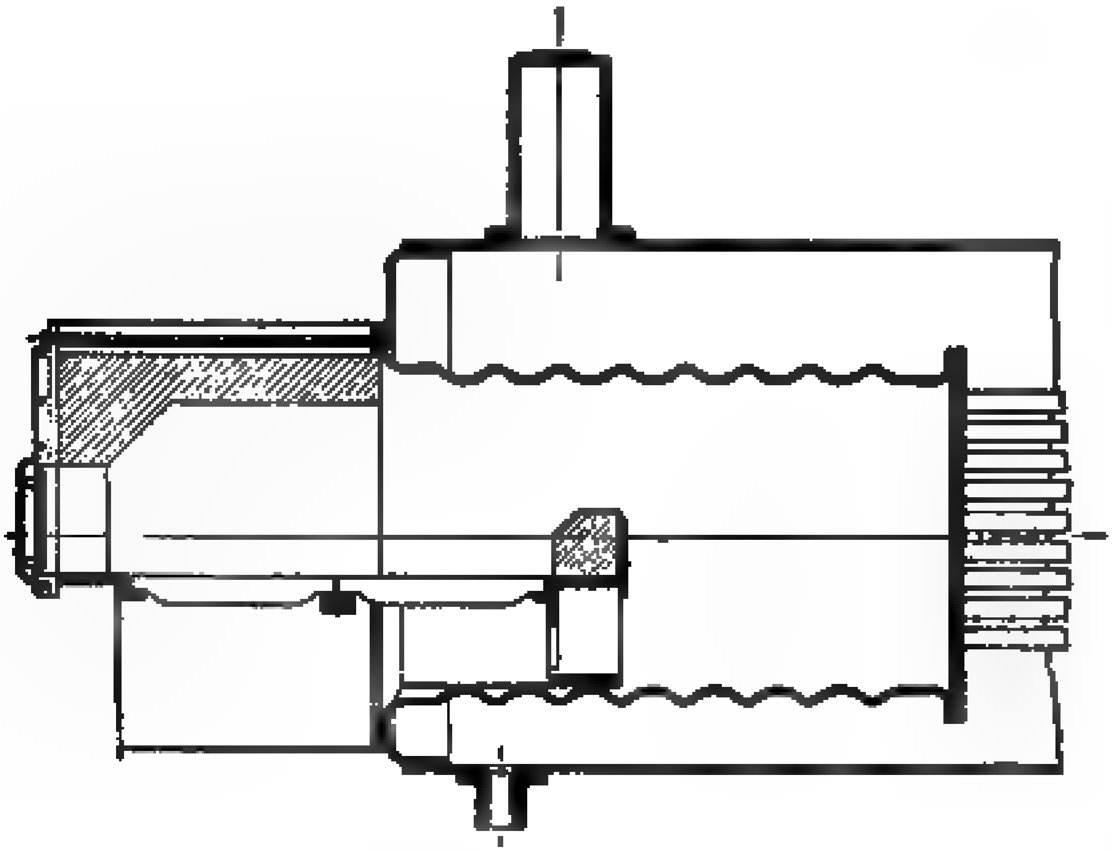
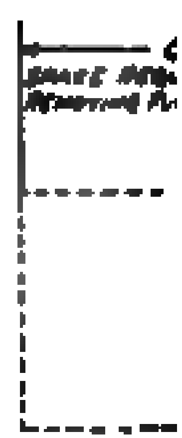


FIG. 1,424.—Sectional view showing Buckeye-mobile extended furnace construction.

FIGS. 1,425 and 1,426.—Buckeye-mobile boiler with furnace and tube unit removed showing construction.



FIGS. 1,427 to 1,429.—Plan and elevations showing space occupied by a 115 horse power Buckeye mobile.

also affords means of removal from the smoke box should occasion require.

Feed water heater.—The heater is of the closed tubular type, the tubes being accessible for cleaning without removal from the shell.

Feed pump.—The boiler feed pump is belt driven from the main engine. An injector is also furnished as a second feed.

Platform and stairs.—The unit is provided with suitable stairs and railings giving access to all points requiring observation and adjustment.

Lubrication.—A force feed lubricator, driven from the valve gear, supplies oil to the valves, pistons and piston rod packings. All the principal bearings are lubricated from a gravity oiling system. The used oil is gathered at one point, flows through a filter and is returned to the supply reservoir for further use.

Governor.—The governor is of the centrifugal-inertia type.

Safety Stop.—The safety stop is independent of the governor. This stop closes the throttle valve if the engine run a few per cent faster than its normal speed from any cause whatever.

Barring Gear.—A barring device is provided to enable the operator to readily turn the engine by hand.

Relief Valve.—Condensing Buckeye-mobiles are provided with an atmospheric exhaust valve which automatically opens when the vacuum is broken for any cause.

Condenser.—A jet condenser is ordinarily furnished for condensing service but a surface condenser can be supplied when local conditions such as bad feed water, etc., require it. The air pump is belt driven from the engine crank shaft.

CHAPTER 27

STEAM HOISTS

Introductory.—The hoisting engine represents a class of machinery possessing wide utilities, for besides general hoisting work, there are other special duties to which it is adapted. Its development has been brought about chiefly by builders of this class of machinery being brought into touch with contractors, miners, bridge erectors, freight handlers on ships, docks, etc., resulting in a machine well adapted to the requirements.

There are a number of terms relating to steam hoists which are here given and which will be found helpful in understanding the subject.

GLOSSARY

Aerial dump.—An improved type of conveyor having low sides, and open at one end. The form facilitates the handling of material by a cableway.

Boom fall rope.—The rope which operates the vertical movement of the boom of a derrick.

Boom swinging gear.—A device for operating a radial or boom swinging derrick. It consists of a drum, or two spools on which the boom swinging ropes are simultaneously wound, and unwound.

Bull wheel.—On radial derricks, a large wheel at the base of the derrick consisting of a circular flange made of structural T iron to which the boom swinging ropes are attached.

Clam shell crab.—A term used to describe a form of bucket used in dredging, excavating, and conveying loose material such as coke, sand, etc.

Cone.—An iron cone shaped cap used in logging operations to assist the log to pass over obstacles when skidding.

Derrick swinging ropes.—The two ropes which control the radial position of the derrick, or by means of which the derrick is made to revolve.

Differential brakes.—Band brakes, in which both ends of the band are pivoted to the rocker but at different distances from the center of the load held, tending to turn the rocker in the direction to apply the brake.

Dock wheels.—Small cast iron wheels fitted to hoists and forming a running gear, for use on docks or places where it is necessary to move the hoist frequently.

Drum.—A revolving flanged cylinder, to which the hoisting rope is attached.

Drum Spring.—A spiral steel spring which pushes the drum out of contact with the friction woods when the friction pressure is released by the friction lever.

Dumping block.—A form of block consisting of a small and large wheel. A piece of chain inserted in the hoisting rope at the right point causes the small wheel to revolve which raises the back end of the skip, and allows the load to *dump*.

Foot brake.—A bank brake for controlling the movement of the drum.

Fixed drum.—A type of drum in use where only single loads have to be handled.

Friction lever latch.—A device consisting of a thumb latch and detent, engaging with serrated teeth in a quadrant, by which the lever is held in any desired position.

FIG. 1,430.—Orr and Sembower double cylinder single friction hoisting engine and boiler specially adapted for pile driving, quarries, railroads, coal yards, docks, and general hoisting purposes. *The principal dimensions are:* cylinders, $6\frac{1}{4} \times 9$; drum, 14×25; drum flanges, 24; boiler, 36×84; 72, 2 in tubes; weight hoisted single rope, 4,000 lbs; weight pile driving monkey 2,000 lbs.; horse power, 18. The hoist has bronze castings in drum, double V friction cut gears, steel pinion, and boiler constructed for 125 lbs. working pressure.

Fig. 1,

FIGS. 1,432 to 1,437.—Twin City hoisting engine cylinder dissembled. *The parts are:* 1, cylinder casting; 2 and 3, heads; 4, steam chest cover; 5, valve stem stuffing box gland; 6, steam pipe flange.

Guard bands.—Protecting wrought iron bands fitted over the gear wheels to prevent the rope or any obstruction getting in the teeth of the gear.

Locking levers.—On double friction drum engines designed for handling a boom derrick. When the friction lever is moved, for the purpose of throwing the drum into gear, the pin on the end of the friction lever presses against the curved part of the oscillating lever and releases the foot brake. When the lever is moved back, the operator places his foot on the foot brake for a moment, and the oscillating lever drops back into its original position, thereby locking the brake on the drum.

Loose drum.—A drum, free to revolve on its shaft, and which is thrown in or out of engagement with the driving shaft by means of a clutch.

Main hoist rope.—The rope which raises or lowers the load.

Operating levers.—Devices for operating the brake, friction, etc., in hoisting. They are usually assembled together and conveniently located for the engineer.

Orange peel grab.—A term used to describe a dredging device which when opened assumes the shape of an orange peel divided into four parts.

Outhaul rope.—In logging operations, an auxiliary rope, worked by a separate drum. It is used to carry the skidding rope out into the woods, which, by former methods had to be done with mules or men.

Post brakes.—A type of brake in use on reversible, mining, and hauling engines.

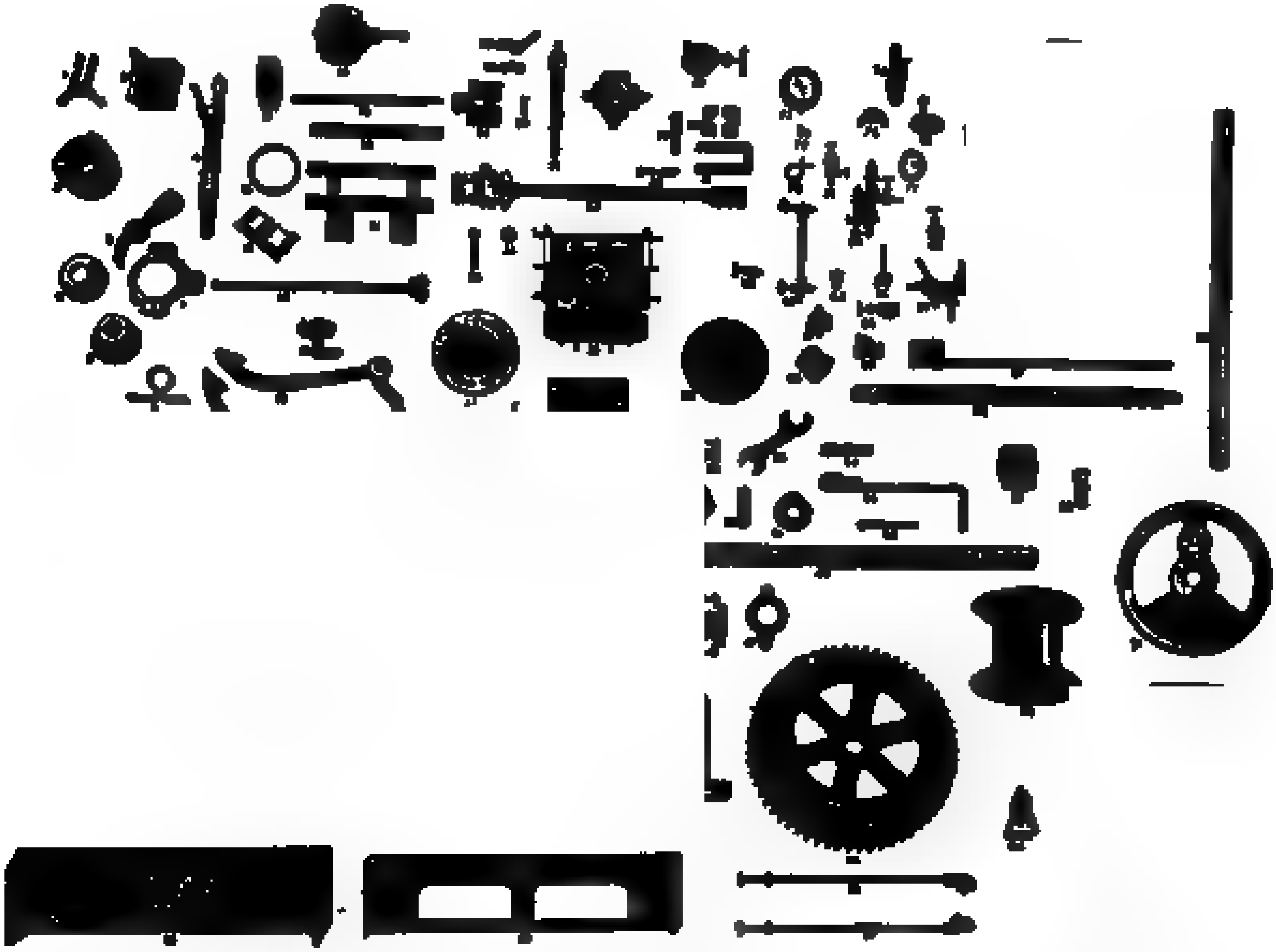
Radial ribs.—Cooling ribs placed outside of the drum friction surfaces to assist in carrying off the heat generated.

Shrouded ratchet ring.—A ring having ratchet teeth around its circumference and bolted to the drum flange. The outer edge of the ring is provided with a shrouded flange, thereby preventing the teeth being broken by the pawl, or the pawl slipping off.

FIG. 1,438.—Street double cylinder horizontal engine suitable for coal elevators, sugar mills, ice houses, dredge boats, logging pull boats, driving conveyors, etc.

Side stands.—Supporting frames usually of T shaped section and designed to carry both the drum and crank shaft bearings.

Skip.—An iron box with a bale holding from one to ten tons of ore or rock; used for hoisting and running between guides, or in inclined shafts to run on a track.



FIGS. 1,439 to 1,540.—Parts of National hoisting engine: 1, clutch winch yoke; 2, boiler bracket; 3, clutch for clutch winch; 4, clutch winch operating lever; 5 clutch winch pawl; 6, plain eccentric; 7, eccentric strap; 8, hub eccentric; 9, pipe holder on boiler; 10, socket wrench; 11, top drum pawl; 12, lower drum pawl; 13, valve rod stand, 14, eccentric rod and end; 15, slide valve; 16, piston ring; 17, piston; 18, throttle valve; 19, piston rod; 20, top guide bar; 21, bottom guide; 22, throttle valve rod support; 23, cap for above support, 24 cross head; 25, cross head key; 26, throttle valve operating handle; 27, three way cock; 28, angle valve for steam pipe; 29, connecting rod brass box; 30, connecting rod key; 31, connecting rod strap; 32, connecting rod bolt, 33, connecting rod, 34, guide bar block; 35, guide bar bolt; 36, cylinder; 37, plain cylinder head; 38, cylinder cock; 39, steam chest cover; 40, claw drum cap; 41, drum pawl pin; 42 crank cap; 43, long bearing drum cap; 44, long bearing clutch cap; 45, threaded shaft collar; 46, drum spring with clamps complete; 47, drum spring clamp; 48, drum shaft; 49, claw; 50, friction key; 51, friction collar; 52, friction pin, 53, friction lever;
gland wrench, 56, steam chest stuffing box gland; 57 steam chest
linder head; 59, piston rod brass gland; 60, check valve; 61, blow off
, water gauge; 64, oil cup; 65, angle valve; 66, injector strainer, 67,
e; 69, sight feed lubricator; 70, syphon for steam gauge; 71, plain
steam gauge; 73, steam gauge; 74, safety valve, 75, arch for hand hole
te; 77, gasket for hand hole plate, 78, brake band; 79, adjusting end
e toggle; 81, stud jaw for brake band; 82, double jaw end for brake
amp; 84, foot brake weight, 85, brake shaft collar; 86, foot brake
t brake operating lever with step; 88, crank shaft; 89, pinion; 90,
, winch head, 93, friction drum; 94, friction block, 95, friction gear;
rod with end; 97, right hand side valve rod with end; 98, open end
ece; 100, right hand side frame; 101, ash pan; 102, ash pan circular
end piece.

Sluing drums.—Two pony drums located on one shaft, and especially adapted for the operation of radial or swinging boom derricks.

Snaking and loading machine.—In logging, a hoist with rigging so arranged that the operations of hauling and loading may be carried on from separate booms independently and at the same time.

Stiff leg derrick.—A form of derrick fitted with wooden braces instead of guy ropes for support.

Tail sheave.—A pulley used in the "pull boat system" of logging.

Winch heads.—A short hoisting drum or spool having curved flanges, and attached to an extension of the drum shaft.

Classes of Hoists.—The great variety of hoisting work, calls for different kinds of engine designed to meet special conditions. They may be classified in several ways:

1. With respect to the steam features; as,

    a. Single cylinder;
    b. Double cylinder;

2. With respect to the transmission of power; as,

    a. Direct drive;
    b. Geared transmission.

3. With respect to the drum arrangements; as,

    a. Fixed drum;
    b. Loose drum;
    c. Single drum;
    d. Multi-drum;
    e. Cone drum.

4. With respect to the conditions of service; as,

    a. General service;
    b. Coal hoisting;
    c. Boom swinging;

d. Logging;
e. Cableway;
f. Mining.

FIG. 1,541.—Simple hoisting engine with direct drive. There are two slide valve engines with cranks at right angles, hence, the engine will start in any position. This type is adapted to high speed hoisting, as in unloading coal at docks.

The foregoing is intended to give an idea of the varieties of hoisting engines in general use.

The Simple Hoist.—In its simple form, the steam hoist consists of one or two engines and a boiler mounted on one base plate, a hoisting drum mounted on the engine shaft, as shown in fig. 1,541. A friction brake is provided so that the drum may be prevented turning when desired, The power of the engine is applied in hoisting by means of a cable, fastened to the drum; the cable is wound upon the drum in hoisting the load, and unwound when it is lowered.

FIG. 1,542.—Usual type of boiler for steam hoists. View showing furnace, and lower tube sheet.

The Boiler.—Steam is supplied to the engine by an upright tubular boiler, usually having through tubes, and supported with the engine on a single base plate. A view of the boiler showing interior of the fire box and tube sheet is shown in fig. 1,542.

There should be ample heating surface and grate area for the engine so that the work may be done with easy firing; the engineer, then, may operate the engine without too frequent interruptions.

The table of proportions given on the next page represents the practice of the builder of the Twin City hoists, for general work and pile driving where one drum is sufficient.

An injector is used to feed the boiler; the other fixtures consisting of safety valve, steam and water gauges, gauge cocks, and blow off valve.

In some cases there is a feed pump operated by the engine; this is a good method of feeding the boiler when a feed water heater is provided.

Twin City Double Cylinder Steam Hoist
One Friction Drum with Boiler

Hoisting capacity single line	Rated H. P.	Cylinder sizes		Drum sizes		Boiler			Shipping weight approx.
		Bore	Stroke	Diam. body	Length between flanges	Diam. shell	Height shell	No. 2″ flues	
4,000	20	6¼	8	12	23	36	75	60	6,300
5,000	25	6¼	10	14	26	38	84	68	7,600
6,500	35	7	10	14	26	40	90	85	7,800
9,000	45	8½	10	14	27	42	96	92	9,500
14,000	65	10	12	16	32	[illegible]	102	150	17,000

FIGS. 1,543 to 1,552.—Twin City boiler, hood, grate, and lugs. The boiler is the vital part of a hoisting engine as it limits the operation. Accordingly special attention should be given to secure the correct proportions of fire box, grate areas and heating surface, so that it will furnish ample steam for the work for which it is intended. These proportions are given in the table above.

The exhaust pipe terminates in a two way cock* with one branch connected to the smoke stack, and the other open to the atmosphere. Steam from the engine, then, can be exhausted directly to the atmosphere, or through the stack; in the latter case, a strong draught is induced through the furnace. Usually an independent jet blower is provided to accelerate the fire when the engine is not in operation.

*NOTE.—Care should be taken to ascertain that the two way cock gives full opening, and if not a larger size should be used otherwise an unnecessary waste of power will take place.

FIG. 1,553.—A single cylinder hoisting engine with geared transmission. For ordinary work it is well adapted, but for heavy duty two cylinders are preferable.

The Engine.—Steam hoists are usually fitted with ordinary slide valve engines. For light service a single cylinder is sufficient, as in fig. 1,553, but for general hoisting work two cylinders

FIG. 1,554.—A two cylinder hoisting engine. The cranks being at 90 degrees there are no dead centers, thus giving better control with heavy loads.

are preferable. They are attached to the bed plate on each side of the boiler, as illustrated in fig. 1,554, having cranks at right

FIG. 1,555.—Buffalo rocking valve type hoisting engine. *In construction*, the cylinder, steam chest, and guides form one casting; the piston rod has a tapered end which is secured to the piston by a nut.

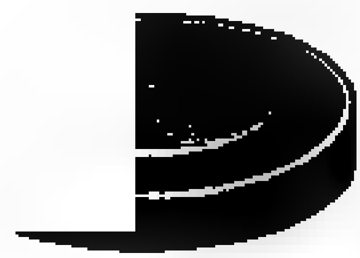
angles; with this sequence of cranks there are no dead centers, hence, the engineer has a more flexible control.

Figs. 1,556 to 1,561 show the type of cylinder in general use with heads, steam chest and pet cocks cover removed. A flange, and stud bolts project on the exhaust side for attaching the cylinder to the base plate. The cross head guide, consisting of a single rectangular bar is attached to projecting piece on the stuffing box of the cylinder head. The cylinder drain cocks are operated simultaneously by a lever conveniently located for the engineer.

Power Transmission.—For light loads, and high speeds, as on coal wharves, engines are sometimes made with a direct drive, as shown in fig. 1,541, that is, the crank is on the same shaft as the drum. With heavy loads, it is desirable to place the drum on a separate shaft as shown in fig. 1,553 and reduce its speed

by means of spur gearing, thus allowing the engine to make several revolutions to one of the drum. By this means a large cylinder or high steam pressure is avoided.

Drums.—Of the many types of drum there are two general classes: the *fixed*, and the *loose* drum. The fixed drum is found



FIGS. 1,556 to 1,561.—Hoisting engine cylinder covers and pet cocks before being assembled. The flange on the side and bolts projecting from the steam chest are for fastening the cylinder to the base plate.

on some of the simpler machines designed for that class of work which requires only the operations of lifting or lowering, or where only single loads have to be handled.

The loose drum is the type mostly used and is so constructed that it may be thrown in, and out of engagement while the engine is in motion, by the action of *friction rings*.

Fig. 1,564 shows the principle of operation.

The drum D, is thrown into gear by a slight endwise movement on its shaft A, produced by turning a lever attached to the screw P, whose end

FIG. 1,562.—National double cylinder, single friction drum hoisting engine without boiler, with ratchet, pawl, foot brake and winch head. This style engine is adapted for general contracting purposes where steam or compressed air is already supplied. It is used to advantage on docks, in tunnels, aboard vessels for handling cargo, for small mines, etc., being very compact and readily made portable.

FIG. 1,563.—National single cylinder, double friction drum hoisting engine without boiler, with ratchets, pawls, foot brakes and winch heads. This engine is also provided with wide face fly wheel which can be used to drive a belt for running pump, etc.

is in contact with a hardened *thrust pin*. The action of this pin is best seen in the sectional view, fig. 1,565. When the engineer turns the friction lever toward him, the thrust screw, which has a left hand thread, pushes the thrust pin against a collar key, which works in a slot cut through the shaft. The collar key, together with the collar, is, in turn, pushed against the drum, forcing it into contact with friction blocks to prevent any

Fig. 1,564.—Sectional view of loose drum. The parts are: A, shaft; B, friction blocks; C, friction flange, D, drum; E and F, friction rings; H, winch head or spool; J, J, bearings; K, gear wheel; P, thrust screw; S, drum flange, T, ratchet teeth.

Fig. 1,565.—Sectional view of friction control on loose drum. When the friction lever is turned counter-clockwise, the left hand friction screw pushes the thrust pin to the left. The pin in turn pushes the collar key, collar, and drum in the same direction causing the latter to engage with the friction blocks.

rotation on the shaft. These blocks consist of sections of hard wood B, bolted to the gear wheel K, which is keyed to the shaft. The adjacent flange on the drum has inclined rings E, F, which register with the bevel on the friction blocks, the endwise movement given the drum by turning the friction lever, bringing the rings into frictional contact with the blocks, thus preventing the drum turning on the shaft since the latter is keyed to the gear wheel K.

When the friction lever is turned clockwise to release the drum, the latter is pushed to the right out of engagement with the friction blocks by the spring M, allowing the drum to revolve freely on the shaft. The loose drum and parts are shown in figs. 1,566 to 1,573 before being assembled.

Loose drums are usually fitted with band brakes in order to reduce the wear on the friction blocks. This is especially de-

Figs. 1,566 to 1,573.—Loose drum and friction control before assembly. The parts are: A, shaft; B, friction blocks; C, block retainers; D, drum; E and F, friction rings forming a V groove on drum flange; H, winch head or spool; I, collar; J, thrust collar; L, thrust pin; N, washers.

sirable for long descents, as in tall office building construction, the wear and heat generated being excessive.

Sometimes radial ribs are placed at the friction end of the drum to assist in dissipating the heat, by presenting additional surface for radiation, and inducing air currents.

The pile driver illustrates a class of work in which the loose drum is most valuable. Instead of loosening the monkey from the rope and having

to re-attach the two before the monkey can be lifted again, the rope remains permanently fastened, and the monkey, in falling, simply unwinds the rope, the drum revolving on the shaft in a reverse direction, and being again thrown into gear after the blow.

Foot Brakes.—When there is a winch head or spool, as H, fig. 1,564, it is necessary to have a band brake if the spool is to be used at the same time that the load is hanging on the drum.

In fig. 1,574 is shown the usual type of brake which consists of a metal band lined with hardwood segments, and embracing the external circumference of the drum flange.

The brake is operated by the foot, and, as shown, is of the differential type in which both ends of the band are pivoted to the rocker, but at different distances from the center of the brake shaft, so that the strain brought by the load held, tends to turn the rocker in the direction to tighten the brake. By crossing the ends of the band at the rocker, the rotation of the brake shaft is limited, so that the foot lever, when released, cannot be lifted by its counterweight above a fixed and convenient position.

FIG. 1,574.—Band brake for holding the drum with load when disengaged from the friction. The metal band lined with hardwood segments tightly embraces the circumference of the drum when the brake lever is pressed down.

Double Drum Boom Swinging Hoist.—The multiplication of drums is a feature in the development of the hoisting engine, adapting it to service of a universal character, such as the working of a radial derrick where frequently several operations

have to be performed at the same time. An example of hoist adapted to this class of work is shown in fig. 1,575.

The forward drum is for hoisting the load, and the other for raising the boom. Between the drums and winch heads are two spools for the derrick swinging ropes. With this arrangement, the operations of hoisting the load, raising and swinging the boom can go on simultaneously.

The winch heads are secured to the shaft by a sliding key having a sleeve upon which the swinging spools revolve.

FIG. 1,575.—Double drum hoist with boom swinging gear for operating a radial derrick. The machine is so constructed that the operations of hoisting the load, raising and swinging the boom may go on simultaneously.

Friction adequate for the full power of the engine is used between the swinging spools and the winch heads. The frictional contact is produced by cams placed between the swinging drums and the frame of the engine; on the other side are anti-friction collars to take up the thrust.

The operating cams are connected by links to the end of a lever, which is carried on a shaft extending across to the side where the engineer stands, and where the vertical detent lever for operating the swinging device is placed

In operation, a rope is wound on the bull ring of the derrick and secured to the same in the center, as shown in fig. 1,576. Each of the two ends is fastened to one of the two swinging spools, enough rope being wound on each to give the required amount of motion to the bull wheel. By throwing the lever forward, it engages the friction clutch on that winch, causing it to rotate and wind in the rope, which turns the boom in one direction, the other winch meanwhile paying out the rope.

HOISTING ROPES

FIG. 1,576.—Base of a radial derrick, showing the bull wheel and boom swinging ropes, also the ropes for hoisting and raising the boom.

The construction of the cams permit this, as the same movement which engages one friction, disengages the other. By throwing the lever backward, the friction of the rear spool is engaged, and the forward spool released. Thus, the boom can be turned in either direction.

The cams operating the frictions are so constructed that when the operating lever is in its central position, there is enough frictional contact in the swing spools to keep the ropes taut; therefore, when one spool is winding in the swinging rope, the other has sufficient friction to prevent the rope, which is paying out, overrunning.

The brake and friction levers for the drums are shown in figs. 1,577 and 1,578.

The friction lever of the rear drum, as shown in fig. 1,577, has an extension piece 1 with a pin, which moves upon the curved surface of an oscillating lever C. The latter has a notch at the lower end, engaging with a pin extending from the side of the foot brake lever. When the friction lever is moved, for the purpose of throwing the drum into gear, the pin on the end of the friction lever presses against the curved part of the oscillating lever and releases the foot brake.

FIG. 1,577.—Lidgerwood brake and friction levers with safety attachment. When the friction lever A, is moved to throw the drum into gear, the pin I, on a projection of the friction lever presses against the curved part of the oscillating lever C, and releases the foot brake.

The friction lever of the front drum, as shown in fig. 1,578, is arranged with a thumb latch and detent engaging with serrated teeth in a quadrant by which the lever is held in any desired position. The drum is thus securely locked, so that hoisting may go on without any further attention to this lever.

When the lever is moved back, the engineer places his foot on the foot brake lever for a moment and the oscillating lever drops back into its

original position, engaging with the pin on the foot brake lever, and thereby locking the brake on the drum. It will be seen that this drum, with the boom hanging from it, is locked fast by the brake, the pawl being usually applied as an additional safeguard.

Directions for operating: The engineer first throws in the front friction drum lever sufficiently to hoist the load. He is then free to handle the boom, either raising or lowering it with the rear drum.

FIG. 1,578.—Friction lever on hoisting drum; a latch and quadrant hold the drum in any position.

When lowering the boom, he should do it both by the friction lever, and a slight pressure on the foot brake. This prevents the oscillating lever falling into place and locking the brake; it also allows control over the lowering speed of the boom, and as soon as it is lowered far enough, then, full pressure is put on the brake lever, and the friction lever thrown out, when the oscillating lever falls into place and locks the brake fast. This use

of the two levers in combination for lowering the boom provides against the possibility of the foot slipping off the foot brake.

It is important that the engineer keep the safety lever locked to the foot brake, and also the pawl applied at all times when the boom drum is not in use. The only exception to this rule is when working with a short boom and the time required to throw in the pawl is too valuable to lose. In such cases, the safety lever locked to the foot brake performs all that is required of the pawl and ratchet.

FIG. 1,579.—National cone friction, gear actuated derrick swinger. The apparatus consists of two small drums mounted on a steel shaft set in two heavy frames in front of engine being attached to main engine frame with steel screw rods. The swinging is effected by gear A, which is mounted on outer end of front drum shaft, which gear drives friction gear C, gear C, driving friction gear B. Mounted on the same shafts with friction gears C and B are two, friction driven pinions which mesh with large gear mounted on swing drum shaft The friction gears C and B, are operated by heavy steel thrust screws, which are connected to main operating shaft on which shaft is placed in a notched quadrant, convenient to engineer, the main operating lever. A movement of the operating lever in either direction causes the swinging drums to revolve as desired. One of the swinging drums is loose on the shaft, being held fast by notched collar. To take up slack in the swinging ropes, it is only necessary to release the notched collar. The position of the swing drums being outside of the hoisting engine drum flanges allows a fair lead for all ropes. This swinging gear can be attached to any make hoisting engine.

FIG. 1,580.—Logging or skidding engine. The front drum operates the carriage to get the slack or skidding rope into the woods; the second operates the out haul rope; the third the skidding rope, and the two parallel drums on one shaft are for loading the logs.

The use of the safety lever locked to the foot brake is desirable with a long springy boom, in chaining or dogging rock after a blast, at which time it may not be known whether the rock, which it is intended to hoist, is free.

The effect of applying the strain for hoisting the rock is to spring the boom, and if the dogs fly loose, or the chain break, the boom springs back to its normal position, slackening for an instant the boom fall rope, which may throw out the pawl causing the boom to fall.

FIG. 1,581.—Hendrie and Bolthoff first motion hoist; an unusual construction. In this hoist, the cylinders and drum are located on the same side of the cross head, the double connecting rods straddling the cylinders.

The type of hoist as shown in fig. 1,575, is suitable for light service, but for heavy duty with long booms, it is desirable to have both swinging spools mounted on a separate shaft.

Logging Engines and Log Skidding.—In fig. 1,580 is shown an engine designed, as the result of experience with practical lumbermen, to overcome the difficulty in paying out the rope, which was formerly done by mules or men. With a receding drum, the rope is carried out along the cable without difficulty.

FIG. 1,582.—Method of skidding logs. The main cable is suspended from two suitable trees, about 700 feet apart. The carriage travels on this cable in either direction, and is provided with a block for the hoisting rope. The tongs are secured to the end of the hoisting or skidding rope, and fastened to the logs for skidding. With the receding rope the carriage is paid out over the cable, and controls the slack in the skidding rope.

The drums are all of the friction type and can be operated separately, or together, as desired.

The front drum operates the carriage to get the slack, or skidding rope into the woods.

The second operates the overhand rope, the third the skidding rope, and the two parallel drums on one shaft are for loading the logs. The drum nearest the track should be used for loading, which gives the engineer full view of the work.

The method of skidding logs in swamps is shown in fig. 1,582. The main cable is suspended from two suitable trees, about 700 feet apart. On this cable, the carriage travels in either direction, and is provided with a block for the hoisting rope. Tongs are secured to the end of the hoisting or skidding rope and fastened

to the logs for skidding. With the receding rope the carriage is paid out over the cable, and controls the slack in the skidding rope.

Hoisting for Deep Mines.—When mining is to be done at great depths, the problem of hoisting the ore becomes more

FIG. 1,583.—Hendrie and Bolthoff heavy duty second motion double drum hoist with balanced slide valve and hand brakes. This hoist is especially intended for heavy and rapid mine work. The drums are grooved to suit rope used. They are loose on the shaft, having removable composition bushings, and are fitted with Webster, Camp and Lane band friction clutches. This enables the drums to be used independently or for balanced hoisting from various levels, and changes from one level to another can readily be made. Each drum is provided with a chain drive dial indicator.

difficult than that met with in ordinary mines. For instance, the variable dead weight of the rope must be taken into account, that is, when the cage is at the bottom, not only is its weight, and that of the one to be hoisted, but also the full weight of the rope, which diminishes as the cage ascends. The rope on the descending side becomes heavier as the cage goes down, with the result that

it takes a great effort for the engine to start the cage or skip from the bottom, while as the ascending cage nears the top, the descending cage and rope may even more than balance the total weight of the former, so that the engine has no work to do during that period. This variable load, of course, results in poor

FIG. 1,584.—Cylinder of Nordberg convertible air hoisting engine for mine service. The cycle of operation is shown in figs. 1,585 to 1,594.

economy, and prevents the use of compound engines. The difficulty is overcome by means of a *tail rope* of the same size and weight per foot as the hoisting rope.

The two ends of the tail rope are fastened to the bottoms of the two skips and at the bottom of the shaft the rope passes around a sheave placed in guides, or on a carriage, according to circumstances. This arrangement

the air is expanded down to atmospheric pressure and compressed to initial pressure and the engine works at perfect efficiency. *Card D.*—In order to eliminate loops in the indicator card air at atmospheric pressure is admitted to the cylinder as the point of cut off shortens, with further decrease in load. *Card E.*—This shows the last diagram for these conditions when the engine is doing no work and consuming no air. *Card F.*—The hoist is now to be retarded without applying the brakes. The operator moves a regulating lever and the hoist becomes a compressor with gear for changing the point of closure of the exhaust valves, which allow the escape of air up to any point of the stroke. *Card G.*—By movement of the regulating lever from neutral position the amount of air rejected from the cylinder is decreased and the amount compressed increased. *Card H.*—A larger compressor card is obtained. *Card I.*—A full capacity card is obtained. *Card J.*—By moving the regulating lever to its extreme position an auxiliary valve is actuated and communication established between the ends of the cylinder when the piston passes dead center and both exhaust valves are closed. This causes the compressed air filling the space behind the piston to flow over to the other side with an increase in pressure, a larger card and about 50 per cent higher mean effective pressure which brings the hoist to a standstill.

balances the hoisting rope, and the load on the engine is, therefore, at all times only the weight of the ore that is being hoisted. This counterbalancing is desirable, aside from the smoothness of operating, because the engine can be better proportioned for the work, resulting in greater economy.

Cone Drum Hoisting Engines.—A development of the single cylindrical drum with two ropes is the double cone drum as shown in fig. 1,597; this is another way of counterbalancing the variable weight of the hoisting rope. The theory of this construction is that as the weight of the rope increases, the radius at which it acts correspondingly decreases, thus equalizing the work of the engine.

Fig. 1,597.—Hendrie and Bolthoff Corliss valve double cone drum first motion hoisting engines. The object of the cone drum is to equalize the work on long hauls by varying the wind radius inversely with the variation in the weight of rope paid out.

The dimensions of the drum must be proportioned for each particular case, and if used under different circumstances, the drum would fail to fulfill the purpose of its design.

For very deep hoisting in order to balance the great weight of rope, the central portion of the drum would become inconveniently large in diameter, therefore this portion is made cylindrical, and is used by both ropes. This arrangement, while it shortens the drum, makes it impossible to balance the weight of rope.

Reel Hoisting Engines.—The reel hoist is often used where it is not the intention to hoist always in balance, and where no

tail rope or compensating device such as a conical drum can be used.

The greatest weight to be lifted by any hoist is when the loaded cage is at the bottom of the shaft, consequently all the

FIG. 1,598.—Hendrie and Bolthoff first motion, Corliss valve double reel hoist for flat rope. post brakes.

rope is off the drum or reel. At this point, however, the reel begins to wind on its shortest radius. As the rope winds on the

FIG. 1,599.—Hendrie and Bolthoff first motion Corliss valve, double reel hoist for flat rope post brakes.

reel, the total load decreases while the leverage of the rope on the reel increases, thus keeping the load on the engine nearly

FIG. 1,600.—Nordberg duplex Corliss hoist (built for Inverness R. & C. Co.); side view showing parallel motion post brakes and their suspension, also in the center the Nordberg four gear reverse operated by steam thrust cylinder. Capacity of hoist, 10,000 ft. of 1½-inch rope; 27 — cylinder 34 — 34 × 72; maximum rope pull 42,000 lbs.; 34 drums 10 ft. in diameter

FIG. 1,601.—Nordberg inclined type, four cylinder hoist; front view showing construction of reels, brakes and engine frame. The post brakes, clutches, throttle and reverse are all operated by steam thrust cylinders.

uniform when lifting one cage unbalanced. Moreover, the rope always leads straight to the head sheave instead of at a considerable angle as occurs when winding on a drum. This makes a compact arrangement which is an advanatge in some cases.


RUNNING OVER
OPERATING PLATFORM
WRIST PLATE

FIGS. 1,602 and 1,603.—*Continued.*

in a position corresponding to the movement of the operating lever. In other words, the entire mechanism has followed the movement of the operator's lever and as soon as he ceases movement, the mechanism becomes locked in that position so that the operator has the same control over it as though it were directly operated by hand. Brakes, clutches, reversing gear and throttles, where necessary are operated by this device. ***Safety devices.*** In addition to th
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FIG. 1,604.—Nordberg hydraulic or oil operated thrust cylinder.

Reel hoists are used mostly in the western part of America. As a rule, hoisting is done there from different levels, one car often being hoisted from one level while the car on the other deck may come from another level; under these conditions the hoisting cannot be done in balance.

Capacity of Hoisting Engines.—The horse power required to raise a load at a given speed is equal to

$$\frac{\textit{gross weight in lbs.} \times \textit{speed in feet per minute}}{33{,}000}$$

To this there should be added from 25 to 50 per cent for friction, contingencies, etc. The gross weight includes the weight of the cage, load, and rope. In a shaft with two cages balancing each thus, the net load is taken.

Limit of Depth in Hoisting.—Taking the weight of a hoisting rope, $1\frac{1}{8}$ inches in diameter, at two pounds per foot, and its breaking strength at 84,000 lbs., it should, theoretically, sustain itself until 42,000 feet long before breaking from its own weight. But taking the usual factor of safety of 7, then the safe working length of such a rope would be only

$$42{,}000 \div 7 = 6{,}000 \text{ feet.}$$

If now a weight of three tons, which is equivalent to that of a cage of moderate capacity with its loaded cars, be hung to the rope the maximum length at which such a rope could be used with the factor of safety of 7, is

$$6{,}000 - \frac{6{,}000}{2} = 3{,}000 \text{ feet.}$$

The limit may be considerably increased, by using: 1, special steel rope of greater strength; 2, a smaller factor of safety, and 3, taper ropes.

CHAPTER 28

PILE DRIVERS

A pile driver consists essentially of *a hoisting engine, boiler, derrick, monkey or hammer, and rigging, all assembled on a foundation of heavy skids, or on a scow.*

The derrick consists of an elevated structure having a pulley at the top over which passes a cable from the hoisting engine drum to the monkey or hammer, the latter working in vertical guides. In operation, when the pile is in position under the monkey, the latter is hoisted by the engine, till near the top of the derrick, when the operator releases the drum friction allowing the monkey to fall and deliver a blow to the pile.

The sinking of the pile depends on the character of the ground into which the pile is driven.

FIG. 1,605.—View of pile driver for land pile driving showing construction of derrick and arrangement of hoisting engine and rigging. For marine use the assembly is mounted on a scow.

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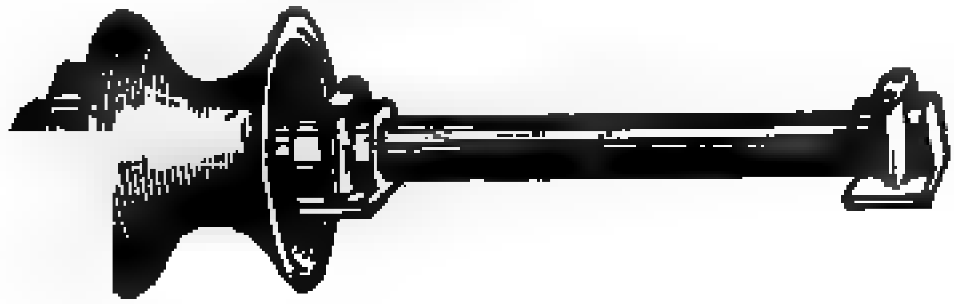
FIG. 1,606.—National portable pile driver for land work mounted on transverse wooden rollers for single motion shifting.

Fig. 1,605 shows the general construction of a pile driver, the type of hoisting engine, mounting, and rigging being plainly shown. For marine use the assembly is mounted on a scow.

lengthwise on a long wooden roller. The wooden rollers are heavily banded at the ends and arranged for moving the derrick sideways. This arrangement permits moving in any direction to suit the convenience of the work.

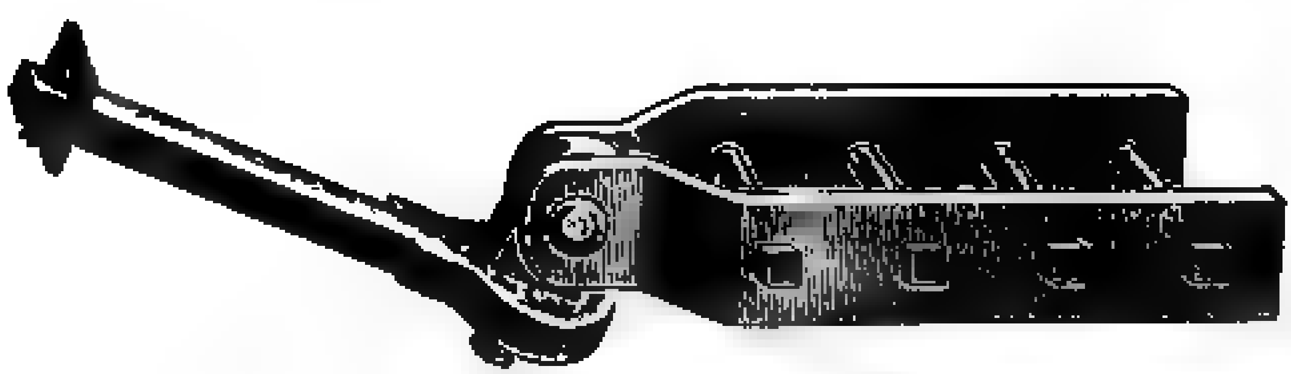


FIGS. 1,608 and 1,609.—Pile driver fittings. Fig. 1,608 roller spool and axle; fig. 1,609 head block with center oiling axle compression grease cups.

FIG. 1,610.—Bucyrus locomotive pile driver with leaders in travelling position. **This machine** has been designed for the use of railroads whose requirements demand a machine of sufficient power and strength to drive concrete and other piles with rapidity, and with sufficient propelling power to obviate the necessity of an attendant locomotive. The Bucyrus driver is provided with a large locomotive boiler and has double cylinder engines of about 300 *h. p.*, which are readily capable of driving the machine with a load of 250 tons, inclusive of its own weight, at a speed of 25 to 30 miles an hour. It may be equipped with a hydraulic turn table, capable of lifting the driver from the track and turning it end for end in ten or fifteen minutes. ***Abstract of specifications.***—Engines, 11 × 12 ins.; boiler, locomotive type, 54 ins. by 15 ft. 9 ins.; boiler working pressure, 175 lbs. per sq. in.; car length, 40 ft.; journals, 5½ × 10 ins.; leaders, 47 ft. long, air break cylinder, 10 × 12 ins.; driving distance ahead from center of forward wheels, 20 ft.; driving distance at right angles from center of track, 21 ft.; driving distance on turn table swung across track, 32 ft.

The following table gives pile driver proportions as recommended by the Lidgerwood Co.

Derrick				*Monkey*		
25 foot	suitable for	1,000	to	1,200	pound
30 "	" "	1,200	"	1,500	"
35 "	" "	1,500	"	1,800	"
40 "	" "	1,800	"	2,500	"
45 "	" "	2,500	"	3,000	"
50 "	" "	3,000	"	3,500	"
55 "	" "	3,000	"	4,000	"
60 "	" "	3,500	"	5,000	"



FIGS. 1,611 to 1,614.—Pile driver fittings. Fig 1,611, pile band; fig. 1,612 monkey or hammer for use with friction drum engines; fig. 1,613, one pulley bracket for face of mast; fig. 1,614, boom fitting with shank to fit pile driver hammer.

For quick work, the following are suitable proportions of engine, boiler and hammer.

Engine	Weight hoisted single rope, lbs.	Boiler dimensions in inches		Weight of monkey lbs.
		Diam.	Height	
5×8	2,000	32	75	1,250
6¼×8	3,000	36	75	1,600
6¼×10	4,000	38	81	1,800
7×10	5,000	40	84	2,500
8¼×10	8,000	42	90	3,000
9×10	9,000	48	102	3,500
10×10	10,500	50	102	4,200
10×12	12,000	53	102	5,000
12×12	16,000	56	120	6,500

The construction of hoisting engines is explained in detail in chapter 27.

CHAPTER 29

STEAM HAMMERS

A steam hammer consists essentially of a heavy weight connected direct to a piston arranged to operate in an inverted vertical cylinder with suitable valve gear to give proper control, the cylinder being mounted on an A frame which contains the guides for the hammer, and the assembly being placed over an anvil.

The original hammer as invented by James Nasmyth was single acting, operating simply by gravity, the function of the steam being to lift the hammer for each succeeding fall.

The first improvement was made by Rigby, who took the waste steam exhausted from the lower side of the piston to the upper side and so imparted some slight pressure in the descent. It was a stage between the early and the present hammers. In the latter, high pressure steam is admitted above the piston to impart a more powerful blow (compounded of *velocity*×*mass*), than is obtainable by gravity; hence they are termed double acting hammers.

In the modern hammer, steam is used to raise and may also be used to drive down the hammer. By means of the valve system, steam is admitted below the piston to raise the hammer and to sustain it while the metal to be forged is placed on the anvil.

To deliver a blow, the steam is exhausted below the piston, and the hammer is allowed to fall by its own weight.

To augment the blow, live steam may be admitted above the piston to assist in driving it downward.

To deliver a gentle blow, the exhaust steam below the piston may be retained to act as a cushion.

Blows can be delivered at any point of the stroke, quickly or slowly, lightly or with full power of the combined weight of the hammer and force

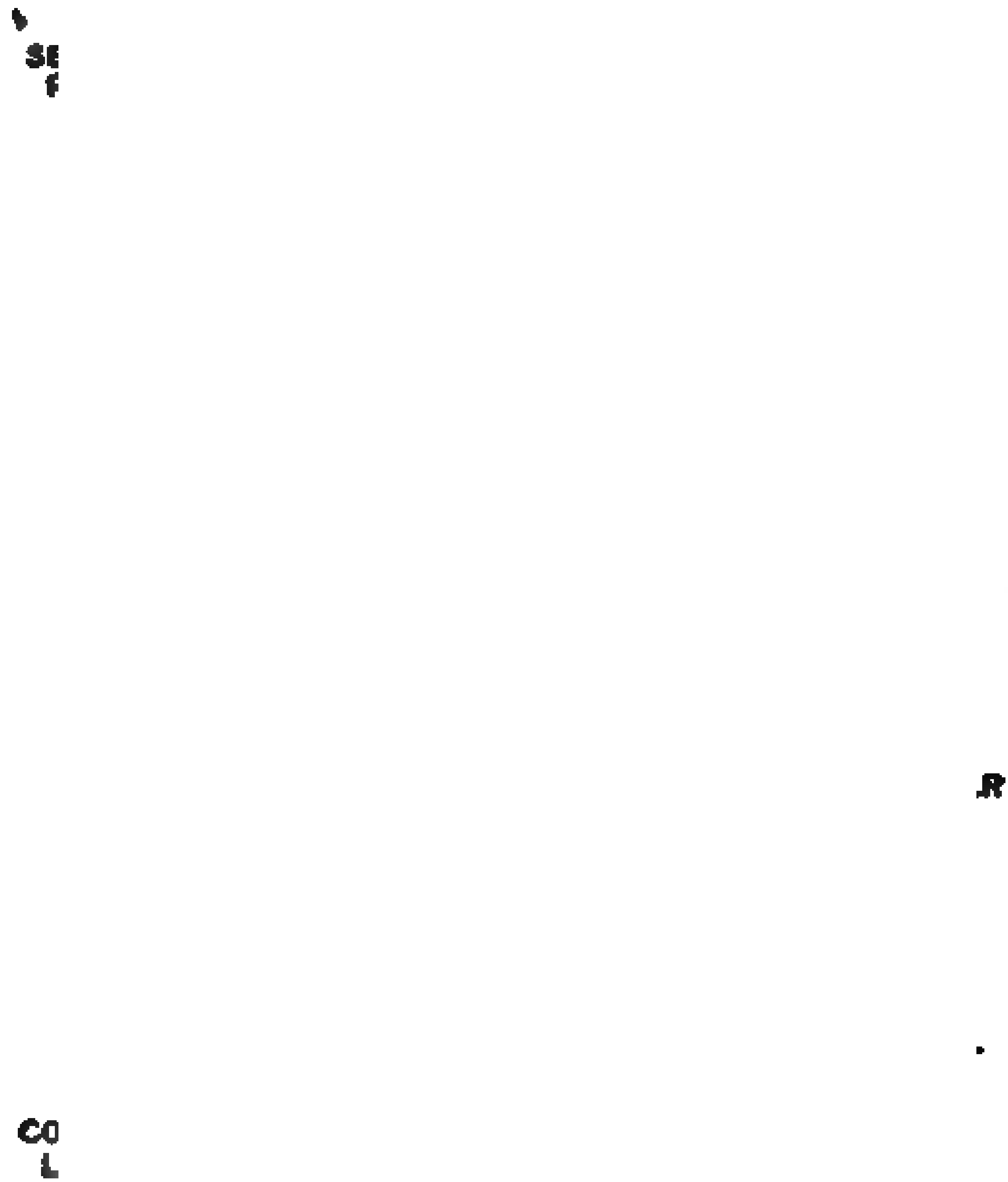


FIG. 1,615.—Sectional view of steam hammer with direct connected self-centering lever.

of steam pressure. The machine may also be used as a vise or squeezer. The various construction details of modern hammers are shown in the accompanying illustrations.

They are rated according to the effective weight of the piston and hammer head and range from 100 lbs. to 80 tons.

The principal difficulties met with in construction are those due to the severe concussion of the blows, which very sensibly shake the ground over a considerable area, accordingly, very heavy foundations are required.

The Valve Gear.—Piston valves are generally used in preference to ordinary slide valves. The periods of admission of steam are under the control of the operator, so that the length of stroke and the force of the blow are responsive to his manipulation of the operating lever.

Many hammers can be set to run automatically for any given length of stroke.

In order that the hammer may reproduce the movements which the operator gives to the control lever, some type of self-centering valve gear is essential. In this type of gear the valve is displaced from its neutral position each time the operator moves the control lever, and is brought back to its neutral position by the self-centering gear, the latter receiving its motion from the hammer.

1. By direct lever connection, or
2. By cam movement.

Fig. 1,615 is a sectional view of a hammer having direct connected self-centering lever.

In operation, the throttle being open, if the control lever A, be moved downward, it will displace the valve in the same direction through link D, admitting steam at the top end of the cylinder and driving the piston and hammer downward. During the movement, link C, being pivoted to the hammer, moves D, downward and with it the piston valve, the latter shutting off steam to the upper end of the cylinder and cushioning the exhaust at the lower end, thus "steam locking" the hammer when it has reached a position corresponding to the position of the control lever.

The essential parts of the gear are very clearly shown in the illustration.

Fig. 1,616.—Sectional view of steam hammer with self-centering cam motion.

Fig. 1,616 shows the cam type of hammer which is virtually the gear as in fig. 1,615 except that the self-centering lever is replaced by a self-centering cam and connecting link as shown.

Hammer Capacity.—It is impossible to rate the capacity of steam hammers with respect to the size of work they should handle, as so many factors enter into the problem that any rate given for one condition would not apply to all.

For making an occasional forging of a given size, a smaller hammer may be used than for manufacturing the same size

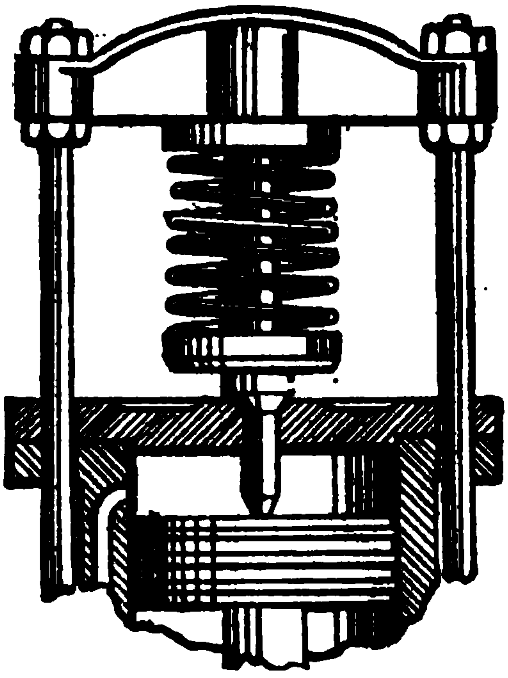


FIG. 1,617.—Erie automatic safety stop. It is attached to the top of the cylinder as shown by the cut. The pin projects through the center of the cylinder head to a point below the top steam port, and is ground in place, making a steam tight joint between pin and head. In case the rod should break, the piston flying upward will not break the cylinder or cylinder head, as the force will be absorbed by striking the projecting pin, causing all shock to be transmitted through the yoke to the side rods shown in cut. The lower ends of these rods are keyed to lugs projecting from either side of the cylinder. Usually the spring will wholly absorb the shock; if not, the worst that can happen is the breaking of two easily replaced side rods.

pieces in large lots. As an illustration, for forging a 6 inch piece, such as a pinion or a short shaft, a hammer of 1,100 pounds capacity would answer, but if the general work run largely to the same size, a 1,500 pound hammer should be used. On the other hand, for forging 6 inch axles a 7,000 or 8,000 pound hammer would be required to ensure maximum output.

For general blacksmith work the following table of sizes may be followed, with modifications according to conditions:*

Diameter of stock	Size of hammers	Diameter of stock	Size of hammers
3½ inches	250 to 350 pounds	5 inches	800 to 1,000 pounds
4 inches	350 to 600 pounds	6 inches	1,100 to 1,500 pounds
4½ inches	600 to 800 pounds		

FIGS. 1,618 and 1,619.—Two types of steam hammer illustrating the difference between single and double hammers.

Boiler Capacity.—For operating steam hammers the size of boiler depends upon the number of hammers in the equipment and the service. It may vary from one boiler horse power for each 100 pounds falling weight, as in a forge where many hammers are in use, to three horse power per hundred pounds falling weight where a single hammer is installed at hard service.

*NOTE.—As recommended by the Niles-Bement-Pond Co.

FIG. 1,622.—View of Niles-Bement-Pond single hammer giving names of parts and showing foundations. The latter consists of two pieces of masonry or concrete which support the bedplate and need only extend below frost line, and a separate one of timber for the anvil extending beyond the others on each side to gain more bearing surface. The anvil pier is made separate from and deeper than the other piers, to reduce the effect of concussions due to the hammer blows. All the piers should rest on solid ground, or if in marshy places, upon piles or timber platforms. Should the anvil settle and require readjustment, it can be readily reached by removing the earth at either side.

FIGS. 1,693 and 1,694.—Typical cross sections of main frame of Buffalo standard guide or combined hammers. Fig. 1,693 top of guides; fig. 1,694 bottom of guides.

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*NOTE.—Erecting Niles-Bement-Pond hammers. *To set single frame hammers:* Place the hammer over the anvil. Let the dies come together. Plumb the piston rod and adjust the anvil or bed plate, as most convenient, until the die faces match exactly; then insert the anvil keys and screw fast to the foundation timbers with woodscrews. *To erect double frame hammers:* Set the anvil perfectly level, being sure that all parts have a good bearing. The die faces should come together so as to bear all over and not on one or two points. Place the bed plate over the anvil, level carefully and bolt down. Place the ram upon the anvil with both dies keyed in. Erect the frames. Bring the guides together against the ram and bolt to bed plate. Bolt on the cylinder and insert the piston. See that taper hole in ram and taper end of piston rod are perfectly dry and free from oil or dirt. If sent connected they need not be disturbed. *To set valves:* Place the latch handle central on its rack and ram at half stroke. The valve should then be central with steam ports and laps equal; if not, adjust by screwing or unscrewing valve stem connection. When properly adjusted the steam will hold the ram down on the anvil or up against the bumpers by placing the latch handle at the bottom or top of its rack. A satisfactory packing is one of the manufactured kind, made to suit diameter of rod and box, as well as depth.

*NOTE.—Starting and maintenance (Niles-Bement-Pond hammers). *To start the hammer:* First see that piston rod is driven in ram sufficiently firm to lift it. This can be done with a heavy bar or sledge. Then by steam, strike several hard downward blows, with hot metal or lead between the dies, in order to more securely jam the taper end of the rod into the ram. Do not allow the ram to strike the spring bumpers until it is secure. The taper pin through the ram does not hold the piston rod and ram together, but is intended to prevent the rod flying out should it become loose. The real hold is the taper fit of the rod in the hole of the ram. Ordinary care should be taken in driving the anvil taper keys not to force them harder than necessary to hold the parts together securely, as such treatment, together with expansion of the parts when hammer is in service, may cause the anvil cap to crack in corner of die notch. Oil cylinder and working parts. Do not use tallow, but good lubricating oil. Open the drip cock. Place the latch handle at top of rack and turn on very little steam. If the valve will not drop freely at first, pull it down by hand until the cylinder becomes well warmed. Close the drip and practice the effect of different positions and motions of the handle with very little steam, until familiar. Be careful not to hammer upward or to strike the dies together carelessly when cold. See that the dies have good bearings in anvil and ram; bearings at the corners or center only are likely to cause breakages. *The anvil* must be level and die faces bear on each other over their entire surface. When the top of the ram is flush with V mark on the guide, the piston head is striking the bottom of the cylinder; the anvil should then be raised or a higher die used. At night, when work stops, always leave the drip open and the handle at the top of rack. This will prevent freezing. *Examine the packing rings* in the piston every three months. To *get at them* remove the buffers and the piston stuffing box packing. Raise the ram until the piston head rises out of the cylinder. *To remove piston from ram:* Raise the ram to the top of stroke. Wedge a steel pin in the center hole in its under side, against the bottom of the piston rod; then allow it to drop upon the anvil. The pin will drive the piston out. *To remove main valve:* The lower part of

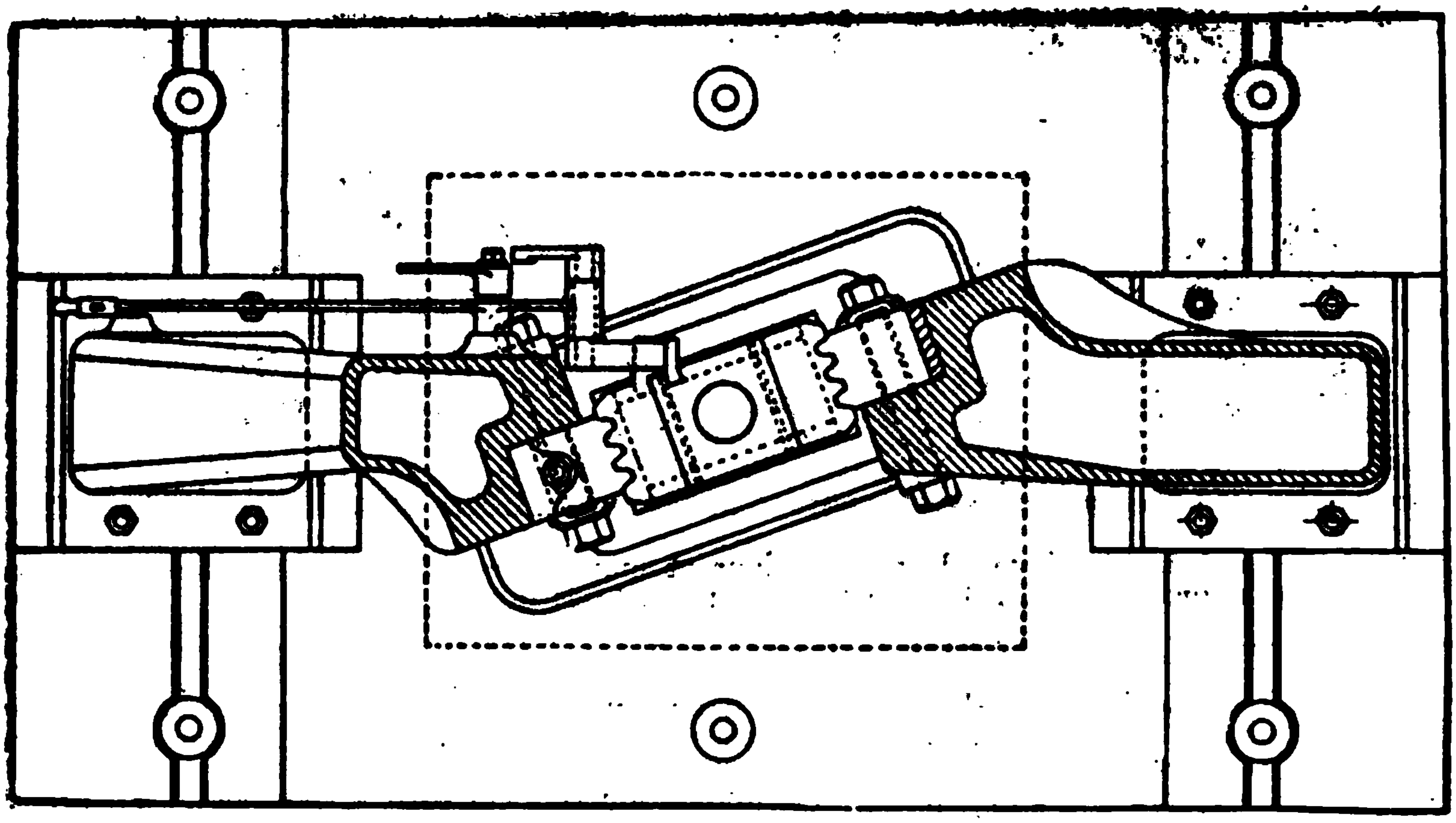


FIG. 1,625.—Cross section of small double frame hammer, showing the hammer head set at an angle with the columns, a construction allowing the use of straight line dies without twisted faces, and making the hammer suitable for general blacksmith work.

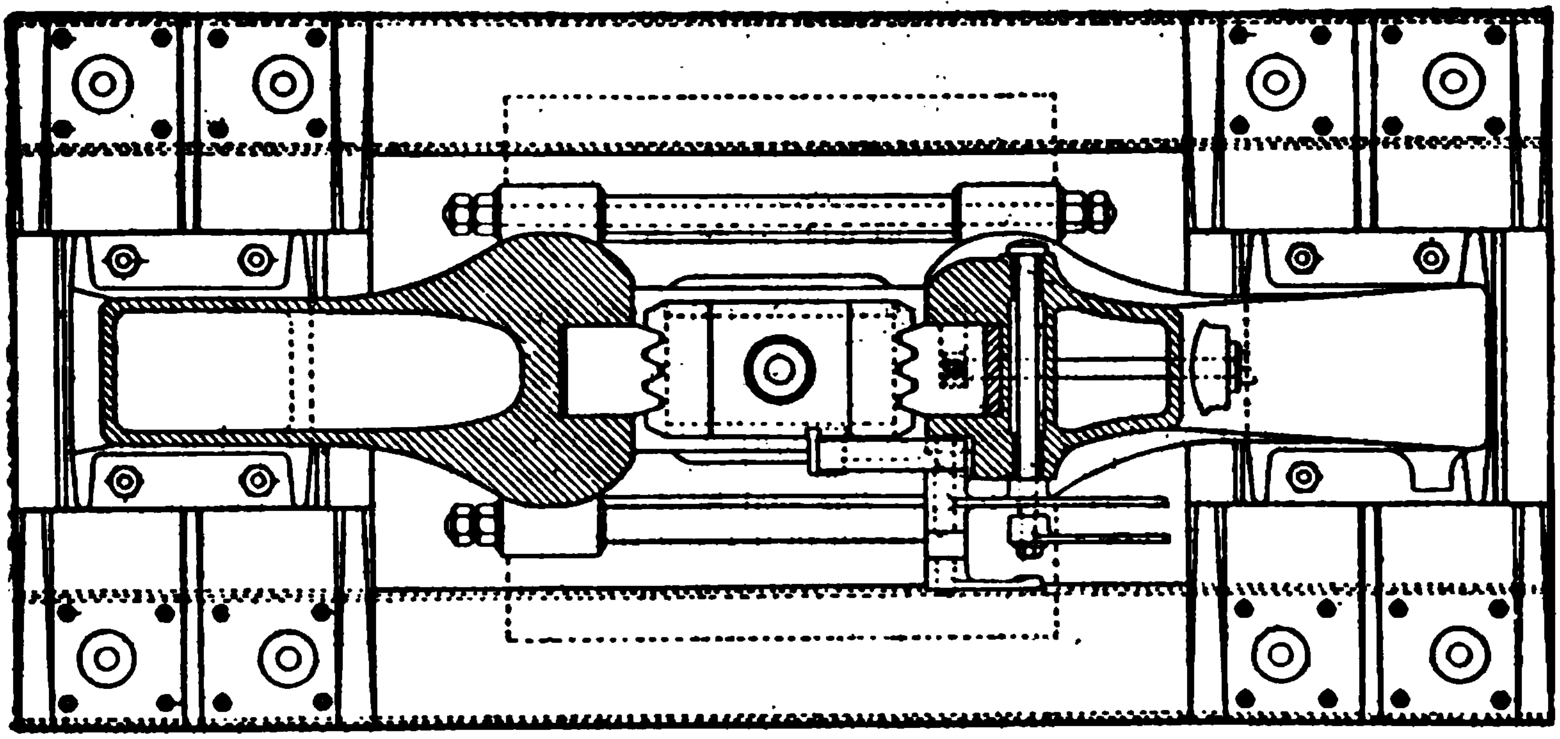


FIG. 1,626.—Cross section of large double frame hammer. On this hammer work is confined to the making of forgings, and in this process forgings are only worked crosswise of the die.

NOTE—*Continued.*
main valve is about one-sixty-fourth of an inch larger than upper part, to insure a slight downward pressure of the valve. When removing the main valve, the liner or bushing must be taken out at the same time, on account of the valve proper being somewhat larger in the lower portion. Do not try to force valve through the casing. ***The steam pipe*** should be well jacketed. Keep the machine well oiled, but use no tallow. ***The exhaust pipe*** should incline downward from the hammer, if possible, or a drain pipe be connected with it to draw off the water. *No cock in the drain pipe.*

The capacity of a steam hammer or its rating is the weight of the ram and its attached parts, such as the piston and rod. The steam pressure behind the piston is not considered as far as the rating is concerned. For example, a 1,000 pound hammer has reciprocating parts of that weight.

***Rule.**—Multiply the area of the largest cross section to be worked by* 80, *if of steel, or* 60, *if of iron, and the product will be required rating of the hammer in pounds.*

Example.—The capacity of a hammer for working steel billets 5 inches square would be determined as follows: $5\times5=25$ and $25\times80=2,000$, which is the rating of the hammer in pounds. A hammer rated according to this rule is an economical size to use, although it can, of course, be employed for heavier work.

Hammers should be operated with steam at a pressure of from 75 pounds to a more efficient pressure of 100 pounds per square inch. They also may be operated by compressed air, in which case the operating valve should be fitted to suit

For average conditions the boiler horse power can be determined approximately as follows:

***Rule.**—Divide the rated capacity of the hammer in pounds by* 100, *and the quotient will be the boiler horse power required for continuous operation.*

Example.—If a hammer be rated at 2,000 pounds, the boiler horse power would be $2,000\div100=20$.

The rule also applies in cases where the hammer is not used continually, by estimating the amount of idle time and making suitable allowance, but the boiler capacity must not be reduced to such an extent that there is a decided diminution in the pressure during the working period.

The capacity of most of the board drop hammers in use varies from 800 to 1,500 pounds; the steam hammers found in drop forging plants usually range from 2,000 to 5,000 pounds capacity, for handling average work. It does not seem practicable to build board drops larger than 3,000 pounds falling weight, and where the forgings are heavy enough to require a capacity over 1,500 or 2,000 pounds, steam hammers are usually preferred. The latter type is also preferred in some forge shops for all classes of work. It is generally conceded that the cost of operation and repairs is greater for steam hammers, but the latter has a greater output for a given capacity.

Board Drop Hammers.—This type of hammer is generally considered superior to the steam hammer for producing drop forgings of small and medium size. When the work is heavy and requires a great deal of "breaking down" or drawing, or even when the forgings are light, but have thin sections that cool quickly, thus requiring sharp, rapid blows, the steam hammer will usually give better results than a "board drop."

The power required for operating board drop hammers varies considerably with the nature of the work. Very little power is required at the point of "pick up," if the work be practically "die to die"; but when the work is soft and there is no rebound, a great deal more power is required, as the rolls have to pick up a "dead load" from rest and there is little kinetic energy in the driving pulleys. When there is a good rebound, with the knock off properly timed, the board will be moving upward with considerable velocity when engaged by the rolls, and much less power is required. Seasoned maple boards have proved superior to any other kind for board drop hammers. Paper fiber has been tried with fair results, but at present the cost of this material is too high.

CHAPTER 30

STEAM SHOVELS

There are few engineering problems which contemplate excavation of earth or rock that do not warrant the use of a shovel. In railroad construction they are employed for making cuts and for digging material, which is deposited into cars to make fills and enbankments and for loading gravel for ballast purposes. For large canal excavation and for shipping overburden from ore deposits and loading the ore, they are in common use. There is scarcely a large brick or clay plant, gravel, cement, or rock quarry that does not consider a shovel an essential part of its equipment. These and many other varied fields for which shovels are used, naturally give rise to numerous types, which may be classed:

1. With respect to locomotion, as

   a. Track;
   b. Traction { skids and rollers; wheels; caterpillar.

2. With respect to radial action, as

   a. Revolving;
   b. Swinging.

3. With respect to the lifting gear, as

   a. Chain;
   b. Wire rope.

FIG. 1,627.—The Thew full circle swing steam shovel. The operative mechanism is mounted upon a rotary bed plate or turn table and carried upon a single four wheel truck. The shovel is mounted upon car or traction wheels and is self-propelling. The hoisting, swinging and crowding motions are controlled by independent engines of the double horizontal reversing type.

Fi

Track shovels.—This type of shovel is mounted on one or more trucks and is designed to run on railroad tracks, for it is adapted to such work as making cuts in railroad construction, and for digging material and depositing same into cars.

In excavating material with a steam shovel, it is necessary; first, to crowd the dipper forward, and second, to hoist it through the material.

Fig. 1,627 illustrates in plan and elevation a small track shovel of the full circle

swing type, and shows the mechanism necessary to produce the compound crowding and hoisting motion.

The dipper is suspended by an adjustable arm, hinged to a carriage or trolley moving horizontally along a suitably designed trackway. This enables the shifting of the point of rotation of the dipper so that the

FIG. 1,629—Thew horizontal crowding motion, for handling shallow cuts, removing concrete and macadam, digging clay and shale, and in general contracting work. It is as nearly automatic as can be expected or desired, and renders the machine practicable for cuts as shallow as six to twelve inches. A cranesman is not required. When power is applied the dipper moves forward into the material, enabling the dipper to fill completely in very shallow cuts. In deeper cuts the trolley is moved forward just enough to secure a cut through the face of bank that will fill the dipper when cut is completed. When reaching the end of the stroke, on either the forward or return movement, the trolley engages a trip rod which throws the throttle of the crowding engine into a neutral position. The range of the crowding motion is such as to permit a long forward movement and correspondingly decrease the frequency of changes in position of shovel. Through the use of horizontal crowding the dipper cleans a floor to the grade upon which the shovel is being operated, thus reducing hand labor about the machine, and the number of pit men required.

hoisting force is at all times exerted in the most effective manner, while the crowding force is applied, in a horizontal direction. This type of shovel resembles in a general way the familiar type of locomotive crane, having its operative mechanism mounted upon a rotary bed plate or turn table and carried upon a single four wheel truck.

The shovel swings through a complete circle, delivering the excavated

material at any desired point either at the side or in the rear of the machine. With shovels of this type, sections of track upon which the shovel operates can be readily handled from rear to front by means of chains hung over the dipper teeth.

FIG. 1,630.—Thew shovel; view illustrating prying motion and swivel dipper arm. *In operation*, by the manipulation of the horizontal crowding motion, a powerful prying action is obtained, the operation being similar to that of a man using a crow bar. The teeth of the dipper are forced under the material and the reverse of the crowding motion applied. The result is a prying action sufficiently powerful to break up most hard materials without the necessity of previously "breaking the bond" or loosening it in any way. The dipper arm, on machines having the horizontal crowding motion, is round and retained in the dipper arm sleeve by means of a swivel clamp. When one of the dipper teeth or one side of the dipper encounters an unusual obstacle, the round dipper arm swivels in its socket, permitting a limited lateral motion. At the same time the dipper arm is relieved from severe torsional strains, which would otherwise be transmitted to the boom, while the pull of the hoisting cable is equalized over the entire cutting edge of dipper lip. By means of the clamp, which also provides the swivel motion, the length of the dipper arm can be adjusted to give the proper cutting angles for the dipper and to dig to any desired depth within a range of about 30 inches below the level of the wheels.

A horizontal motion is imparted to the dipper, by its suspension from a carriage or trolley moving horizontally along a suitably designed trackway. The shovel is mounted upon car wheels, set to narrow or standard gauge, or upon traction wheels, for transportation over city streets; is self propelling.

The hoisting, swinging, and crowding motions are controlled by double horizontal reversing engines. Some of the dimensions of a small full circle swing shovel are as follows:

Swing of boom	18′	Crowding engine double	4×5
Height of boom	18′ 6″	Vertical boiler	36×84
Lift of dipper	10′	Steam pressure	10×100 lbs.
Radius of cut	20′	Boiler feed	Injector
Length of crowding	5′	Capacity of dipper	½–⅝ cu. yd.
Hoisting engine double	5×6	Rated capacity per hour	35–40 cu. yd.
Swinging engine double	4×5	Approximate weight	27,000 lbs.

FIG. 1,631.—Semi-portable mounting consisting of rollers and skids.

FIG. 1,632.—Caterpillar traction, a desirable mounting where the machine is to be moved about to any extent.

FIG. 1,633.—Four wheel equalizing trucks. These may be all non-propelling in which case the drag lines move in the same way as when mounted on skids and rollers or two of them may be driven by power from the main engine. On the two smaller drag lines the trucks would be mounted directly under each corner of the base, but on the two larger drag lines, two of them can be mounted on an equalizing beam as illustrated.

Fig. 1,634 shows a track shovel of the swing type. Here, instead of the entire machine turning, the boom carrying the shovel is pivoted which permits radial movement but to a lesser extent than in the revolving type.

FIG. 1,634.—Bucyrus steam shovel for general contracting and railroad work in average materials.

FIG. 1,635.—Bucyrus steel boom showing swinging and boom engine thrusting gears, etc.

Traction Devices.—Fig. 1,631 shows a semi-portable rig consisting of skids and rollers. This forms a standard mounting for dray line work. When mounted in this way, the dray line (later described) travels on a track of plank laid on the ground and is pulled ahead by pulling itself up to its bucket, which in

this case forms an anchor. The caterpillar traction is shown in fig. 1,632. Where the machine is to be moved around to any extent this is an ideal mounting as it eliminates the gang of men necessary to carry plank and rollers, or to handle track sections.

Modifications of Standard Shovels.—The term standard shovels, applies to those machines having dimensions and characteristics which long experience has shown to be most efficient

FIG. 1,636.—Bucyrus swinging and crowding engine. It is a double cylinder center crank reversing engine. The bed plate is a single casting, with bored guides for the main cross head. and rocker connection for the valve stems.

and best adapted for general requirements. Modified forms are often necessary to meet special conditions of service, and the following modifications are frequently employed.

Sewer Trench Booms.—For sewer trench work at considerable depths below the surface, special booms with long dipper handles and special dippers of reduced capacity designed particularly for such conditions are recommended. Because of the increased leverages and proportionately greater strains such booms are of extra heavy construction. Frequently

FIGS. 1,637 to 1,641.—Various types of Thew steam shovel. Fig. 1,637, type O shovel using horizontal crowding motion to excavate shallow sub-grade for paving; fig. 1,638, type O shovel with horizontal crowding motion excavating a through cut in road construction work, fig. 1,639, type O shovel using shipper shaft mechanism and long dipper stick in trench excavation, fig. 1,640 example of high lift work done by Thew shovel equipped with long dipper stick; lift is 20 ft. above level of wheels; fig. 1,641 example of how Thew shovel digs its way down to the level of the excavation.

propelling, its engines being sufficiently powerful to run it on ordinary roads. It will also dig ditches for draining or reclaiming swamp lands.

the machine is counterweighted to compensate for the additional weight of boom.

Clam Shell Booms. — For clam shell operation or for use as a locomotive crane a special boom is provided with auxiliary mechanism for derricking boom and handling second rope of clam shell or orange peel bucket. For railway construction the standard shovel is frequently employed in the manner indicated on the opposite page. For emergency requirements the so-called single rope clam shells may be used with standard booms.

Drag Scraper Booms.—For Type 1 and larger shovels special booms can be furnished with necessary mechanism for handling drag scraper buckets. Counterweighting is essential to secure a proper radius of action.

Jack Knife Booms. — For work on city streets it is at times desirable to reduce the clearance height of boom to swing under trolley

wires. This may be accomplished without interfering with the efficiency of the shovel in general work by having an adjustable section at end of boom as shown on the opposite page.

High Lift Booms.—Local conditions sometimes demand the use of longer booms to secure greater dumping height or radius. Such modifications result in decreased efficiency, loss in stability and hoisting power, reduced operating speed and increased strains on swinging mechanism. They should be avoided if possible.

FIG. 1,646.—Bucyrus double hoisting engine with link motion reverse. The bed plates are single iron castings, with bored guides for the cross heads.

Special Boilers.—When shovels are to be employed in localities having boiler requirements of a special nature, such fact should be specified in placing order.

CHAPTER 31

TRENCHING MACHINERY

The varied conditions met with in excavating work give rise to several types of machine in addition to those classed as steam shovels, and which are properly known as excavators or trenchers. The principal machines of this class are:

1. Drag line excavators;
2. Trench excavators;
3. Ditching machines.

Drag Line Excavators.—Machines of this type, though not properly classed as steam shovels, are closely related, for with the addition of a boom and dipper handle they become steam shovels.

This machine is becoming recognized more and more as capable of doing work which formerly had been done by hand and team labor and in many cases supplements work commonly done by the steam shovel and the dredge. Its broadest field is perhaps the field of irrigation and drainage ditches.

Because of its wide radius of action it can deposit the material far enough from the cut to keep the weight off the banks and prevent caving. Furthermore, it can dig a much better slope than it is possible to make with a dipper dredge or a steam shovel. It is especially efficient in levee or dam building where it may travel parallel to the work and borrow the material from one side.

Experience in Louisiana, on reclamation projects in the West, and elsewhere, has proved conclusively that it will construct as compact and

watertight embankment as can be built by any other method. In like manner railroad fills may be constructed. Here the rehandling of the material and the necessary extra equipment can be done away with. Sand, gravel and clay pits are also being successfully operated by these machines. Inasmuch as it digs below the level on which it stands, rain may occasionally flood the excavation without stopping the dragline, whereas a steam shovel would be "drowned out." Many pits which have been worked down to water, have been re-opened by the dragline by virtue of this fact. Its wide reach, furthermore, eliminates the necessity of frequent moving. Draglines are also being used for placer mines where conditions prevent the economical use of a large gold dredge.

Some of the dragline excavators now in use are virtually adaptations of light hoisting machines capable of giving fair results only under the most favorable conditions and often not even then. This has led to a general misapprehension as to the actual field of a machine built, as it should be, built along the substantial lines of a steam shovel. Effective and economical operation requires both speed and the great strength necessary for continuous operation.

The accompanying illustrations show the construction and operation of dragline excavators.

FIG. 1,649.—Side elevation of Monighan walking drag line excavator.

holds them in this position by means of a brake. The clutch is then released and the machine resumes excavating.

Trench Excavators.—Perfect ditching is reproducing in the ground exactly to form and dimension the engineer's computed ditch plans. These plans invariably call for sloped sides, wide berms, even spoil banks and a smoothed channel true to line

FIG. 1,651.—Bucyrus dragline excavator stripping phosphate 300 feet from digging, to dumping point. A good operator can throw the bucket from 10 to 40 feet beyond the end of the boom depending upon the size of the machine, and the conditions under which it is working. The digging depth does not conform to any fixed rule. It increases with the length of the boom, but varies greatly with the material being dug.

FIG. 1,652.—Bucyrus drag line excavator on caterpillar traction digging small drainage ditches.

and grade. Unless there be no merit in this precision of engineers' ditch plans, these plans should in actual construction without question be reproduced in the ground.

There is merit in the truly planned ditch section. Volume of flow, speed of current, caving of banks, silting of channel,

FIG. 1,658.—Austin excavator beginning ditch showing method of dumping buckets. This type of excavator is adapted to digging nearly all kinds of ditches when supported on a track resting on the surface of the ground, as shown, one rail being placed on each side of the ditch (or roller platform traction). Thus mounted, it easily propels itself forward or backward at the will of the operator. The presence of water in the ditch does not seriously interfere with the yardage capacity. By commencing at the outlet or lower end of the ditch, the water is disposed of to quite an extent, and generally the machine passes over the wet ground comparatively dry shod. Large lakes and extensive swamp areas have been drained in this manner. A ditch left smooth contains no obstructions, and since banks and bottom are true to grade, it is impossible for it to be filled by erosion, as there is nothing to obstruct the flow of silt, and a waterway of maximum capacity is therefore maintained. Excavated in this manner, the earth taken from the ditch is comparatively dry and is delivered at a distance, insuring a berm of practically any desired width, and the waste banks are therefore never returned to the ditch by erosion, nor will the banks cave, as is the case when torn up by dipper dredges that can deliver the waste banks only immediately adjacent to the ditch.

NOTE.—*Perfect ditches.*—Don't fall into the error of assuming that any sort of a channel which will carry water makes a good enough drainage ditch. To produce a perfect ditch the following requirements must be fulfilled: 1, the bottom of the ditch *must be true to grade* and free from roughness. 2, the sides of the ditch must be true and smooth and must slope back from the bottom at an angle flat enough to prevent the earth falling, caving or slipping. 3, the earth below the true planes of the bottom and sides must be left undisturbed and *as firm as nature made it.* 4, the material excavated from the ditch must be *deposited far enough away from the edge* not to work or slide back and not to bring a caving pressure on the edge of the ditch. 5, the spaces between the edges of the ditch and the bottoms of the spoil banks, which engineers call the berms, *must be wide and clean.*

minimum of excavation are all taken into consideration by the engineer and his plan of ditch section is the one that best meets all requirements. To vary from his plans may cheapen excavation but it is at the expense of efficiency of the ditch.

The function, then of a trench excavator is ***to reproduce in the***

FIG. 1,654.—Small sewer trench dug by Parsons excavator at Aberdeen, S. D. The machine is seen at the farther end of the ditch.

ground in one operation the form and dimensions of the engineer's planned trench section. That is to say, a trench excavator digs to template.

A typical trench excavator such as is shown in fig. 1,653, and suitable for digging drainage and irrigation ditches and building

levees, consists of a structural steel "frame" which spans the ditch and propels itself on wheeled trucks (or caterpillar traction wheels) running along the berms. This frame supports the operating machinery and also a bucket guide frame. The bucket guide frame has vertical movement. When hoisted it

FIG. 1,655.—Parsons ditching machine. It consists of a portable boiler and engine, the latter connected by suitable gearing to the digging mechanism which consists of a chain of buckets as shown, traveling around three double sprockets, and having near the top a conveyor for depositing the excavated material to the side of the ditch. An oscillating device permits of making a ditch of any width between 22 and 42 inches, and which consists of a double worm mechanism with reversing gears which deliver the oscillating power at two points on the digger boom. The double engine delivers power through a large fly wheel and expanding clutch direct to the digging buckets. Traction, steering and hoisting power are delivered independently by a sprocket on the crank shaft. Power steering is through worm and quadrant, shifting one lever being the only operation necessary. The hoisting device is operated through a worm gear by one lever and permits the grade to be maintained. Power is applied to the driving wheels at the wheel rim by bull pinions, these pinions being keyed to the differential shaft. The drivers are supplemented by a heavy steel cable mounted on a drum, which is geared to the driving shaft. In case the drivers should slip in soft ground, the cable may be attached to a dead man ahead and traction obtained regardless of the action of the drivers.

rides clear of the ground, and from this high position it can be lowered by degrees to the depth below the surface of the trench bottom. This guide frame is the template, made to the engineer's ditch plans, on which the excavating bucket travels and cuts the ditch exactly to desired shape.

When grade is reached the guide frame is raised to high point, the machine moves itself ahead the width of the bucket and the operation is repeated.

The travel of the excavating bucket is across ditch. It shaves off a thin slice down one side, across the bottom and up the opposite side. It then makes a return stroke, repeating the cut in the opposite direction. The frame carrying the bucket is lowered as the ditch increases in depth, so that banks and bottom are cut to exact specifications. There is no loose earth left in the trench, and the original strata of earth in the banks and bottom are undisturbed and remain perfectly firm.

The machine constructs a trench any reasonable depth, with slope of sides 1 to 1 or 1½ to 1, at a single operation. The waste banks are also constructed at a distance from the trench, and they may be made continuous to serve as dikes, thus increasing the capacity of ditch; or the earth can be delivered to either side, and at various distances.

Ditching Machine.—This type of excavator differs from a trenching machine in that it digs a narrow deep trench or ditch with vertical instead of sloping sides. Its essential features are: 1, a system of digging buckets which operate in the direction of the excavation instead of transversely as in a trench excavator; 2, a conveyor that will deposit the excavated material on either side of the trench at the will of the operator; 3, propelling mechanism. Fig. 1,655 shows a type of ditching machine suitable for digging ditches of considerable depth and having a range of width between twenty-two and forty-two inches.

CHAPTER 32

DREDGES

A dredge may be defined as *a marine or floating excavator*; their construction has much in common with the steam shovels and excavators described in the preceding chapters. There are several types of dredge, designed to meet the varied conditions of dredging; these may be classed as

1. Dipper;
2. Hydraulic or suction;
3. Elevator.

Dipper Dredges.—This type of machine is virtually a floating steam shovel. It has in common with the shovel, the characteristic features of dipper, handle, boom, rigging, etc. The machinery being mounted on a scow or flat bottom hull.

For securing the dredge in position a system of spuds is employed. These consist of heavy timbers (usually reinforced) which are arranged to move through supports, and by means of cable or rack and pinion movement can be let down vertically and stuck into the bottom of the stream, thus anchoring the dredge.

In some cases the spuds are arranged in an inclined position so as to project to the banks or sides of the stream. Fig. 1,660 shows the general arrangement of a dredge with convertible band and vertical power spuds.

FIG. 1,656.—Interior view of Marion 1¼ cu. yd. dipper dredge showing machinery. *The dredge* is equipped with a 55 ft. boom and vertical spuds *The hoisting and* [illegible] is of standard double hitch type, operated by two 8×10 non-reversible engines fitted with a throttle of the [illegible] *The spud hoist,* which consists of three drums [illegible] to independent frictions, is mounted in a special frame and driven from the end of the backing shaft. *The* [illegible] *nery* is an independent outfit consisting of two drums mounted on a long shaft and compound geared to a 6×7 double reversible engine. *The boilers* are of the locomotive type and are fitted complete with the necessary pumps, injectors, valves and fittings. The machinery is mounted on a 75×32×6 steel hull, and the entire structure is covered with a galvanized steel house

FIG. 1,657.—Marion dipper dredge improving Louisiana Canal system.

FIG. 1,658.—Marion dipper dredge, having 92 foot reach with 83 foot boom.

FIG. 1,659.—Marion dipper dredge cutting ditch through timbered country.

Hydraulic or Suction Dredges.—The essential feature of this type of dredge is a pumping plant of considerable capacity, having a suction pipe projecting over the side of the dredge hull and so hinged that it can be lowered and moved radially over the surface of the material to be dredged.

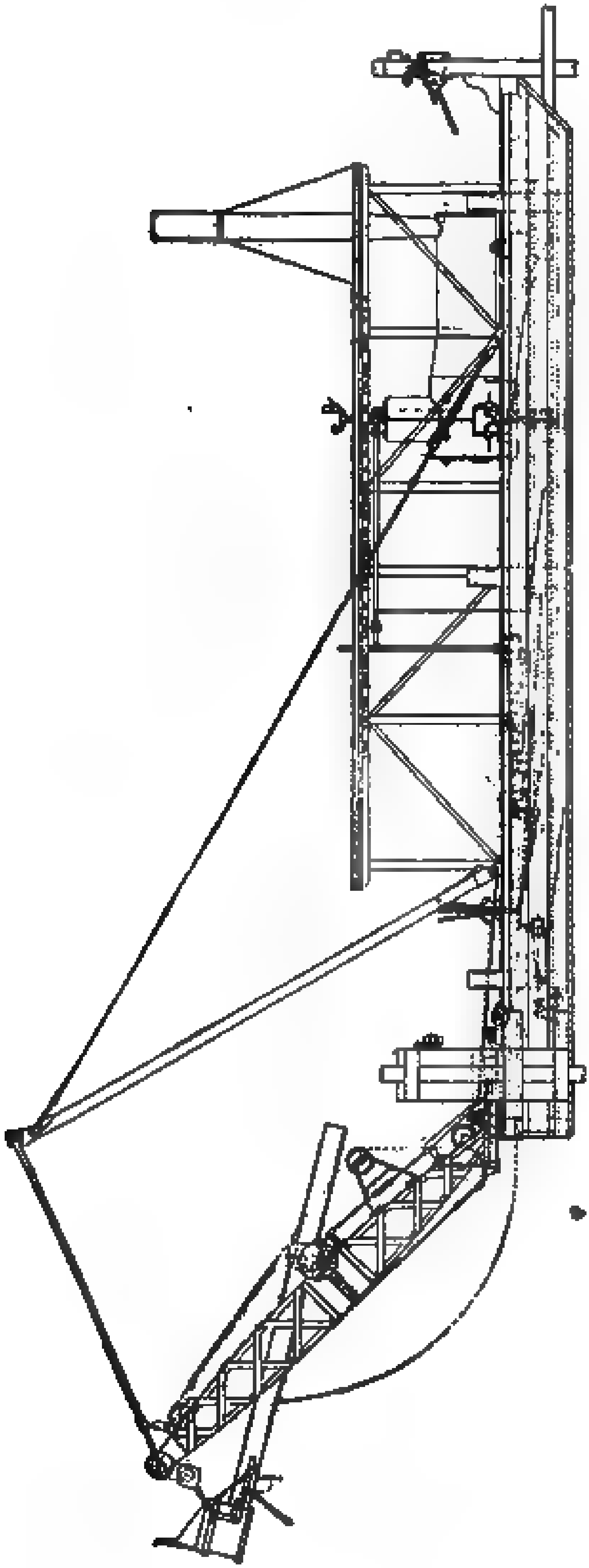
In operation, the dredging pump creates a partial vacuum in the suction pipe which produces a strong velocity of water in same, sufficient to draw in the material and keep it moving; and the pump also produces the

FIG. 1,660.—Fairbank's convertible bank and vertical power spuds. They are operated by means of powerful gears meshing into heavy wide racks attached to the spud members as shown. The drive is by a worm on a horizontal shaft. The method of operation enables the dredge to be moved either fore or aft. The power is supplied by independent reversing valve engines compounded and direct connected to driving gear, which gives the operator complete control. On small machines the spuds are operated by sprocket and chain from hoist engine.

pressure necessary to force through the discharge pipe line to distance desired, and at the same time elevates to reasonable height.

If, occasionally, the material should be found too hard packed, a few jets of water under pressure from a centrifugal pressure pump delivered at the suction intake will help in boiling up the sand so that the pump can draw it in to better advantage.

For general dredging service where all classes of material will be handled it is necessary, however, to use an agitator or cutter to cut and loosen the material, after which the pump will draw it in to the suction pipe.



Here the suction pipe is mounted within a structural steel ladder hinged to the dredge, of suitable length to dredge to the depth required, and of very heavy proportions to stand the strain due to dredging in hard material. The cutter is provided with a series of cutting blades and is mounted on a powerful shaft supported on the ladder and driven through gearing from an independent engine. If properly constructed even shale rock can be dredged.

Usually two spuds are arranged in the stern of the dredge that act as anchors to hold the dredge in position. By means of winging lines to either side of the dredge controlled by a hoisting engine, the dredge is swung from side to side on the spud as a pivot and thus the dredge can also be moved forward as the work progresses and the dredging operation be perfectly controlled.

The hydraulic dredge is very economical in handling sand, gravel, silt, mud, clay, loam, etc., in fact, it can be used in all classes of material except solid rock.

The dipper dredge, elevator dredge, or other types of dredges, are efficient machines but neither can deliver the material except within a very short

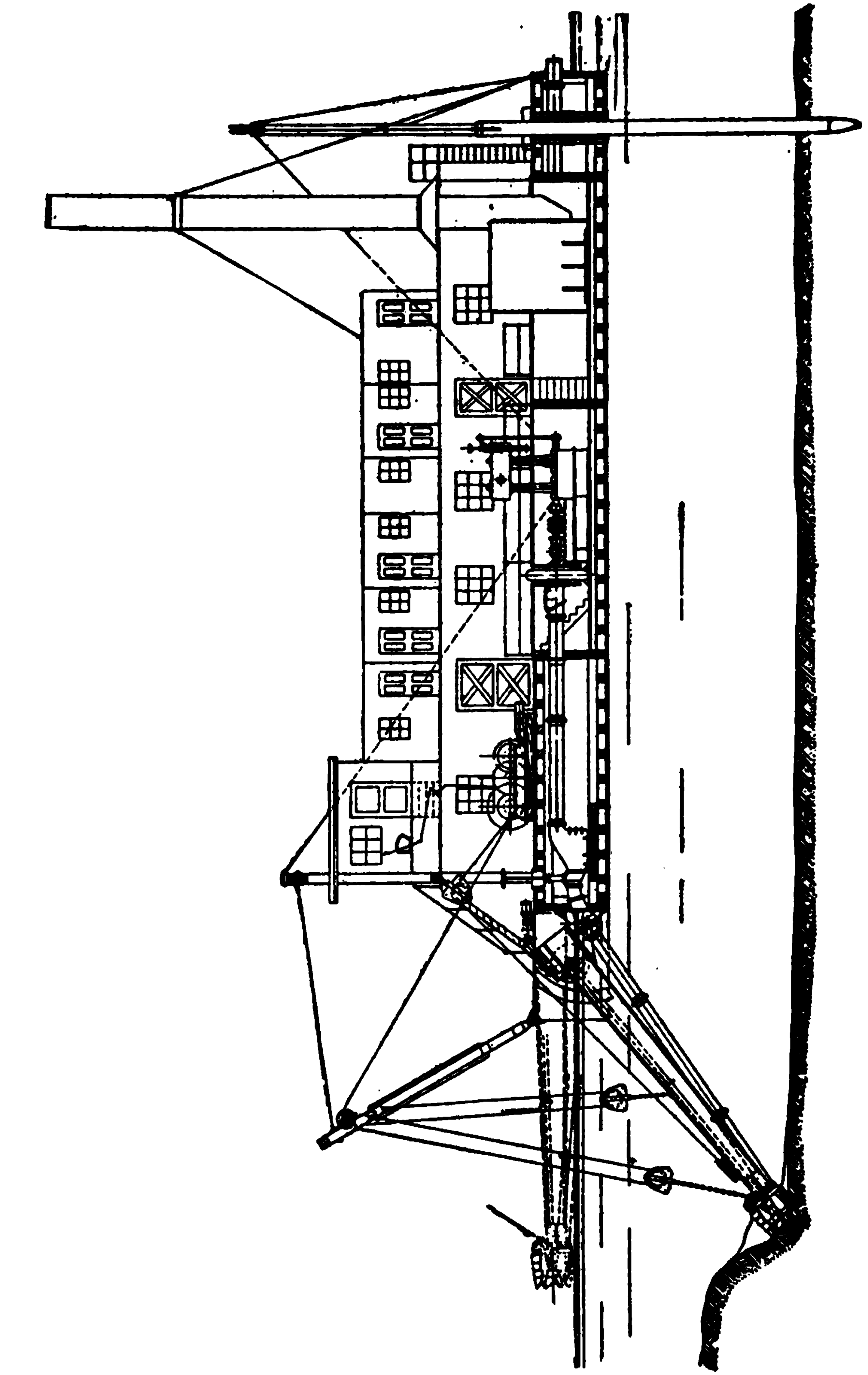


FIG. 1,662.—General arrangement of Norbom hydraulic dredge showing location of boilers, engines, pumps, control machinery, suction pipe and cutter.

radius of the dredging operation. If the material is to be delivered at any distance it must be done by means of, for instance, dump scows requiring tug boats, etc.

The hydraulic dredge not only dredges the material, but also delivers it to just the point desired as regards distance and height, all with one operation; and the cost of the dredge, considering its capacity, is less than any other type of dredge.

A dredge is most efficient when handling the greatest amount of material with the least amount of water.

FIG. 1,663.—Detail of Norbom cutter and suction pipe, showing both in operation under water.

By the use of the rotary cutter and by systematically swinging the dredge and moving forward on the spuds the amount of material fed to the pump can be regulated so that the dredge is carrying the maximum percentage of material constantly.

The operator or lever man has before him a vacuum gauge showing the vacuum in the suction pipe and a pressure gauge showing the pressure in the discharge pipe; the reading of the vacuum gauge when handling material is greater than when pumping water only, and the operator shortly becomes proficient so as to keep the vacuum reading at the point which carries the greatest amount of material without choking the pipe.

delays, the average output may be estimated at from 10 to possibly 15 per cent.

of flexibility,

The power required to drive the dredging pump depends upon the elevation above surface of water to which the material is to be raised, the length of pipe through which delivery is to be made, and character of the material.

In fine sand, silt or mud, a high velocity through the discharge pipe line is not necessary to obtain satisfactory results. In coarse, heavy sand and gravel the velocity must be high.

Overcoming the friction through the discharge pipe, therefore, represents a large portion of the power that must be expended in driving the pump.

FIGS. 1,666 to 1,668.—Plan and elevations of centrifugal pump direct connected to compound engine as used on hydraulic dredge.

In pumping water only, large pipes are used to reduce the velocity and frictional resistance and thus save power. In a dredging pump a high velocity is necessary, however, so that the material may be carried through the pipe without settling and choking the pipe.

For most economical operations as regards power used, the velocity through the pipe line should not be higher than is just necessary to carry the material satisfactorily.

As the frictional resistances decrease with larger sizes of pipe, it follows that with long pipe lines the larger dredge becomes the more efficient in use of power and operating cost per cubic yard handled.

With easily handled material the delivery pipe may be a mile in length or more.

With heavy material, requiring high velocity, the length of the pipe line can usually not exceed 4,000 to 5,000 feet. For longer pipe line the pressure required on the dredging pump becomes too great; there will be difficulty in maintaining the rubber sleeves joining the pontoon pipe, and the wear on the dredging pump becomes excessive.

FLANGE

FIG. 1,069.—Ball joint and hinges on suction pipe of hydraulic dredge.

The practical maximum discharge pressure is from 45 to 55 pounds.

For long pipe line it therefore becomes necessary to use relay pumps; that is, the dredge pump delivers through a certain length of discharge pipe into the suction of a relay pump, which pump then delivers through the remainder of the pipe line.

For high elevations and very long lines several relay pumps may have to be used.

FIG. 1,670.—20 inch Bucyrus hydraulic dredge at work on the Great Lakes.

FIG. 1,671.—20 inch Bucyrus hydraulic dredge digging a canal, showing cutter in operation.

The efficiency of a dredging pump is usually from 40 to 50 per cent. As the pump must be built with large openings to pass all classes of material the efficiency must be largely disregarded. The main requirement is ability to keep going.

For sake of economy dredging pumps of medium or large size are usually directly connected to compound or triple expansion engines.

To better adapt a dredge to the varying conditions of length of pipe line and elevation it is well, with a directly connected pump, to provide several different diameters of impellers so as to enable full boiler capacity and the maximum horse power of the driving engine to be utilized at all times.

Fig. 1,673.—Plan of winding machinery of Norbom hydraulic dredge. The standard outfit has five drums, each with friction discs and band brakes, all operated from a set of levers in the pilot house. The bearings rest on cast iron stands. The winding machinery is sometimes made with six drums, which is especially useful when the ladder is made for independent side swing.

With a long pipe line a large diameter of impeller is required.

If same impeller were used with short pipe line the engine speed would be held down so that the engine could not deliver its maximum power. Therefore with the short line a smaller impeller should be used, enabling the engine to speed up and deliver its maximum power and thus a greater amount of material be handled.

FIG. 1,674.—Parker pontoon pipe coupling being assembled.

FIG. 1,675.—Parker coupling ready for use.

The machinery of a hydraulic dredge may be divided into several units or groups, as

1. Boiler plant;

2. Steam piping and auxiliaries such as condensers, auxiliary pumps, feed water heater, hot well and all such auxiliaries necessary to make a complete plant;

3. Pumping unit;

4. Cutter machinery;

5. Winding machinery;

FIG. 1,676.—Arrangement of operating levers on Norbom hydraulic dredge. These levers can be divided into two groups of five or six levers each, in place of one group as here shown.

6. Hull piping and fittings;

7. General fittings such as sheaves, wire rope, moving bits, spuds, and frame fittings, etc.

Usually some type of shell boiler is used. The dredging pump is of the centrifugal type, sizes 6 to 20 inch being used in the western rivers for dredging sand and gravel for building purposes and other uses. Here usually the sand lies loose and the suction force of the pump is sufficient

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Fig. 1,678.—Bucyrus 3½ ft. gold elevator or placer dredge at work near Nome, Alaska.

to draw it into the suction pipe and obtain a satisfactory capacity without the aid of an agitator.

The dredge for this class of service becomes exceedingly simple, consisting principally of the dredging pump with its driving equipment mounted on a scow; the suction being a pipe of sufficient length to reach to the bottom with a piece of flexible suction hose in same to give necessary flexibility so that the suction can be raised and handled as desired.

The material is usually delivered into a flat deck scow with raised sides, so that the sand is retained and the water flows back into the river. Sometimes these dredge boats are self-propelling and provided with hoppers into which the material is pumped.

FIG. 1,680.—Marion gold dredge used in winter gold dredging, Alder Gulch, Montana.

Elevator Dredges.—This type of dredge is constructed with an endless chain of buckets which travel on sprockets attached to suitable booms, the rig being designed for such service as excavating gold bearing material.

The material is deposited near the center of the dredge, where it is dumped into screens which separate the stones and coarser material from the gold bearing sand. The sand then passes over riffles and gold saving tables and, after the gold is collected, passes on to the tailings pile at the rear.

Fig. 1,680 shows Marion gold dredge used for winter gold dredging, Alder Gulch, Montana.

CHAPTER 33

CONCRETE MIXERS

Concrete mixture is cement, sand, and gravel or broken stone, mixed with water. The quality of a concrete mixture is determined by

1. The quality and proportions of the materials composing it, and
2. Its uniformity.

Hence, the best mixture depends not alone on the quality of material but upon the quality of the mixing.

To properly mix concrete, the elements must be successively subdivided and re-arranged in new relations toward each other. To keep the fine pieces from sifting to the bottom, a pouring action is necessary, combined with a horizontal endwise movement, averaging all parts of the batch.

The essential parts of a concrete mixer are:

1. Charging hopper;
2. Mixing drum;
3. Water tank;
4. Discharge device.

The Charging Hopper.—This should be not over, 36 to 40 inches high, so as to be convenient for either shovel or wheel barrow charging. It should also be wide enough to receive the discharge of an ordinary concrete barrow, and have high wings to keep the gravel from spilling.

FIGS. 1,681 to 1,685.—Various types of Marsh-Capron mixers. Fig. 1,681, mixer with gas engine drive; fig. 1,682, No. 4 mixer steam power; all levers are operated by one man on platform built over the mixing drum; fig. 1,683, No. 0 mixer with 10 h. p. engine; fig. 1,684, No. 3 mixer with hoist and 25 h. p. engine; fig. 1,685, gasoline engine mixer.

Fig. 1,686.—Lakewood charging hopper.

Fig. 1,687.—Grand charging hopper. ***In operation,*** as hoppers B, and C, reciprocate backward and forward, the material that is thrown into hopper B, is forced toward the mixing trough at one move of the hopper, shoving charge I, off into the mixing trough and at the same time "striking off" charge J. At the next movement of the hopper, charge J, is shoved into the mixing trough and charge I, is "struck off." The amount of these charges is determined by the height of steel wire brushes G1 and G2, which are adjustable. The amount of the cement discharged is regulated by a steel slide which passes over the cement feed roll. This slide is drawn out certain distances for different proportions, allowing only the uncovered part of cement roll to fill and discharge. After slide is set for desired proportion it is locked in place by set screw so there is no possible chance for variation in the proportion. By discharging the cement and other materials at the same time we receive a good gravity mix before the material even reaches the mixing trough.

Mixing Drum.—The drum is usually cylindrical and rotates on bearings called trunnion rollers.

The tracks encircle the drum which bear upon the rollers, and support the drum as it turns. At the center is a circular rack to which power is applied through a pinion for driving the drum. At each end of the drum is a head having a central opening, one for charging and one for discharging. The general appearance of a mixing drum as described is shown in fig. 1,691.

FIGS. 1,688 to 1,690. — Miles three compartment charging hopper. *In operation*, cement enters through the center and sand, crushed rock, or gravel from either or both sides. A constant stream of materials are fed simultaneously into the mixing drum, the mechanism being adjustable so that any proportion may be obtained. The feed is controlled by an individual clutch and may be stopped or started regardless of the rest of the machine. Often for various reasons, the shovelers stop for a minute or two. In such cases the feed is thrown out of gear by a clutch lever. Further if it be desired to empty the drum, the feed may be stopped while the balance of the mixer continues until the materials are all discharged. The materials are conveyed or forced into the mixing drum by three conveyor chains with scraper links. These conveyor chains are driven by three sprocket wheels, mounted on a square steel shaft, which prevents any possibility of their slipping and all are traveling at the same speed, and by the adjustments of the steel brushes and the fact that it can be fed from either or both sides, gives the machine a capacity up to one hundred yards in ten hours. The cam or eccentric proportioning discs which are graduated and provided with a positive locking device. *The parts are:* A, chain, B, scraper links for sand; C, scraper links for cement; D, idle sprocket; E, idle sprocket for sand chain; F, idle sprocket for cement chain; G, square shaft; H, sprocket feed drive; J, adjustment for thickness of brushes; K, cement slide; L, brushes for sand or stone; M, proportioning cam for sand or gravel; N, proportioning cam for cement; O, proportioning cam for stone or sand; P, slotted slide for holding proportioning cam; Q, lock nut for double locking proportions; R, short section of mixing drum.

Water Tank.—The water used in mixing concrete is usually placed in an elevated tank containing some device to regulate the flow.

Fig. 1,691.—Lansing mixing drum showing tracks which are cast integral with the head having a flange at each outer edge which serves as guides for the trunnion rollers.

LIFTING BUCKETS

MIXING BLADES

CAST STEEL BRACKETS

WALLS ½" THICK

Fig. 1,692.—Interior of Chain Belt mixing drum showing lifting buckets, moving blades, drive sprockets, etc.

On one type of tank, the amount of water desired is regulated by a float inside of the water tank. This float can be set by the contractor according to the consistency of the batch he desires. By raising the float the flow of water is increased, and by lowering, decreased.

When the desired flow of water is secured, the float can be set and there will be no other adjustments necessary until the contractor wishes to change the consistency of his batch.

The construction of the water tank is very simple, consisting of the float, one check valve, one three way plug valve, and one hand lever by which it is operated.

Fig. 1,693.—Municipal automatic measuring tank for measuring the water for each batch. The supply can be varied according to requirements by raising or lowering the gauge rod projecting through top of tank.

Methods of Discharge.—The two general methods of removing the concrete mixture from the drum is

1. By tilting the drum;
2. By inserting a chute.

FIG. 1,694.—Clover leaf mixer, showing involute curved drum. ***In operation,*** the drum revolves counter clockwise. As the material is carried along it falls of its own weight when it reaches a certain point. In falling the material is doubled over several times each revolution.

FIGS. 1,695 to 1,697.—Lakewood automatic tank. The water tank is filled by attaching a hose to the one-inch pipe connection on the three way valve below, and pulling the operating handle down, when the tank will fill to the top and close automatically. By pushing the handle up, the connection from the supply hose is cut off and the discharge side of the valve is opened, allowing the water in the tank to flow into the mixer. The amount discharged from the tank into the mixer can be regulated by the loop of two inch pipe on the discharge side of the tank by placing this loop in a vertical position if a small amount of water be desired, and by placing the loop in a horizontal position when the full contents of the tank are desired. Any amount between the minimum and maximum can be discharged, by placing the loop at intermediate positions between the vertical and horizontal.

FIG. 1,698.—Milwaukee mixer with batch hopper, engine boiler and run ways. All mounted on a flat car, discharging over the end of the car.

FIG. 1,699.—Smith tilting drum hot mixer on asphalt repair work. In the hot type of mixer a steam blower creates a forced draught from the boiler through the mixing drum. The gases enter the mixer at a temperature well above 600° Fahr. As the aggregate is sprayed from the blades, every surface is dried and heated by immediate contact with the hot fumes. The rapid motion of the particles prevents sooting or burning of the material. The bitumen is introduced after the drying process is complete, by means of the tank on the top of the skip. The output of the mixer varies with the size, character and condition of the aggregate and the temperature desired for the finished batch.

Fig. 1,699 shows a mixer discharging by the first method. Figs. 1,700 and 1,701 illustrate the chute method of discharge.

Power Transmission.—On all mixers the engine furnishes power for turning the drum, and in many cases, provision is made to apply the power for loading and discharging the drum.

Fig. 1,706 shows a chain drive with countershaft for turning and loading the drum.

On the Smith mixer which has a tilting drum, the tilting motion is reversible, permitting discharge of any part of the batch. At the points of greatest travel, the clutches are automatically disengaged. The tilting action is controlled by a revolving screw and traveling nut.

FIG. 1,702.—Ransome mixer with boiler and fixed batch hopper.

FIG. 1,703.—Ransome mixer with boiler and feed chute.

FIG. 1,704.—Ransome mixer with boiler and pivot hopper.

FIG. 1,705.—Ransome mixer with boiler and sliding pivot hopper.

NOTE.—***Proportioning Concrete.*** The following four mixtures will serve as a rough guide to the selection of proper proportions for various classes of work (*Taylor and Thompson*): *1.* ***Rich mixture*** for columns and other structural parts subjected to high stresses or requiring exceptional water tightness. Proportions, 1: 1½: 3. *2.* ***Standard mixture,*** for reinforced floors, beams and columns, for arches, for reinforced engine or machine foundations subject to vibrations, for tanks, sewers, conduits, and other water tight work. Proportions 1: 2: 4. *3.* ***Medium mixture,*** for ordinary machine foundations, retaining walls, abutments, piers, thin foundation walls, building walls, ordinary floors, sidewalks, and sewers with heavy walls. Proportions 1: 2½: 5. *4.* ***Lean mixture,*** for unimportant work in masses, for heavy walls, for large foundations supporting a stationary load, and for backing for stone masonry. Proportions 1: 3. 6. The above specifications give fair average practice. If the aggregate be carefully graded and the proportions are scientifically fixed, smaller proportions of cement may be used for each class of work.

The Power Unit.—The conditions under which a concrete mixer is used are such that the power machinery must be very dependable, since usually it receives little or no care, being entrusted sometimes to an engineer of low grade. Accordingly, the power outfit is of the simplest type consisting of a slide valve

FIG. 1,703.—Chain drive for turning and loading drum of chain belt mixer. The loading mechanism consists of a hoisting drum and clutch as shown. The loader is charged on the ground level directly from the wheel barrows, being filled while the previous batch is being mixed in the drum. As soon as this batch has been mixed and discharged from the drum, the loader is elevated and another full batch emptied into the drum. The loader bucket is raised and lowered by means of a four part line. The cable used is three-eighths inch plow steel, six strands of 19 wires each attached to a hoisting drum operated by a wood friction clutch.

engine and upright shell boiler designed to operate at about 80 lbs. pressure.

Figs. 1,707 and 1,708 show both sides of the Lakewood mixer and give a very good idea of the power equipment. As shown the mixer is at one end of the frame and the boiler at the other, the engine being between the two. The figure also shows clearly the transmission between the engine and boiler consisting in this case of double reduction spur gearing.

FIG. 1,709.—Standard street paving mixer at work.

FIG. 1,710.—Ransome street paver equipped with 20 feet distributing boom and automatic dumping bucket. All controlling levers are brought to a point within easy reach of the man required to operate the machine. Standing on the platform at the right, he raises the pivot hopper, empties the water tank, operates the discharge chute, sends the bucket out along the boom, and controls the point of automatic dumping almost without effort. From the same position he controls the engine, operates the traction clutch, and steers the machine. A two barrow pivot hopper, across which extends a dumping bar to prevent the wheel barrows running back, permits quick charging. For discharging, either a boom and automatic dumping bucket or a distributing chute is provided. The boom is twenty feet long and has a swing of 180°. The distributing chute has a discharge radius of fifteen feet, with intermediate outlet gates, and like the boom has a swing of 180°. This machine will lay from 1,250 to 1,500 sq. yds. of concrete road, 5 inches thick, in a ten hour day. The traction speed of the machine is 4,775 feet per hour, forward and reverse.

ide loader mixer. One

Figs. 1,713 and 1,714.—International concrete mixer (globe type), and section of self-oiling winding drum. The view shows discharge side of the mixer. ***In construction,*** it is mounted on a steel frame and arranged for motor drive by belt. The drive and countershafts are special, not regular design, and may be reversed to opposite end when conditions necessitate changing location of motor. The standard machine is mounted on trucks and is provided with side loader and automatic water tank.

CHAPTER 34

TRACTION ENGINES

Most of the operations in connection with the production of crops can be done better and quicker by the use of steam than any other power. Plowing, harrowing, seeding, harvesting, threshing, grinding, etc., are all regular work for the traction engine. On account of the nature of the work, traction engines are also appropriately called the *agricultural* or *farm engines*.

The use of these machines has made possible the extensive development of the world's agricultural resources that has taken place within the last generation. While apparently tending to displace manual labor, they have in reality had the effect of giving more work and better wages to the working man.

The hard usage to which the traction engine is subjected require that it be built of superior materials, and be of substantial design. It must not only propel itself, and haul its thresher over the roughest roads and fields, over hard and soft ground, but it must haul plows successfully, and withstand the strain of pulling a grader through most obstinate soils.

Since incompetent men, or careless hired help are often placed in charge, it is sometimes subjected to jars, shocks and misusage which other kinds of machinery do not encounter.

A traction engine is a self-propelled machine and consists of:

1. A single, or double cylinder reversing engine;
2. A boiler, usually horizontal;
3. Running gear with differential device on rear wheel;
4. Clutch and transmission gears;
5. Engine and boiler accessories and connections.

FIG. 1,716.—Eclipse engine. A single cylinder throttling engine with heater and cross head pump. The figure shows the control levers, clutch, and shifting eccentric reverse gear.

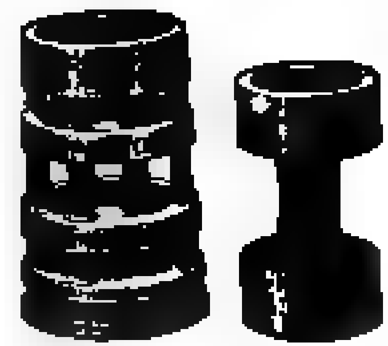
The Engine.—This may be the simple cylinder, duplex, or compound type. In the first instance, a friction clutch is necessary, because the engine might stop on a dead center, and would be difficult to start by hand if in engagement with the transmission gear.

FIG. 1,717.—Reeves two cylinder engine. Both steam chests are connected to the main steam pipe by a T branch; the cranks are at right angles thus avoiding dead centers.

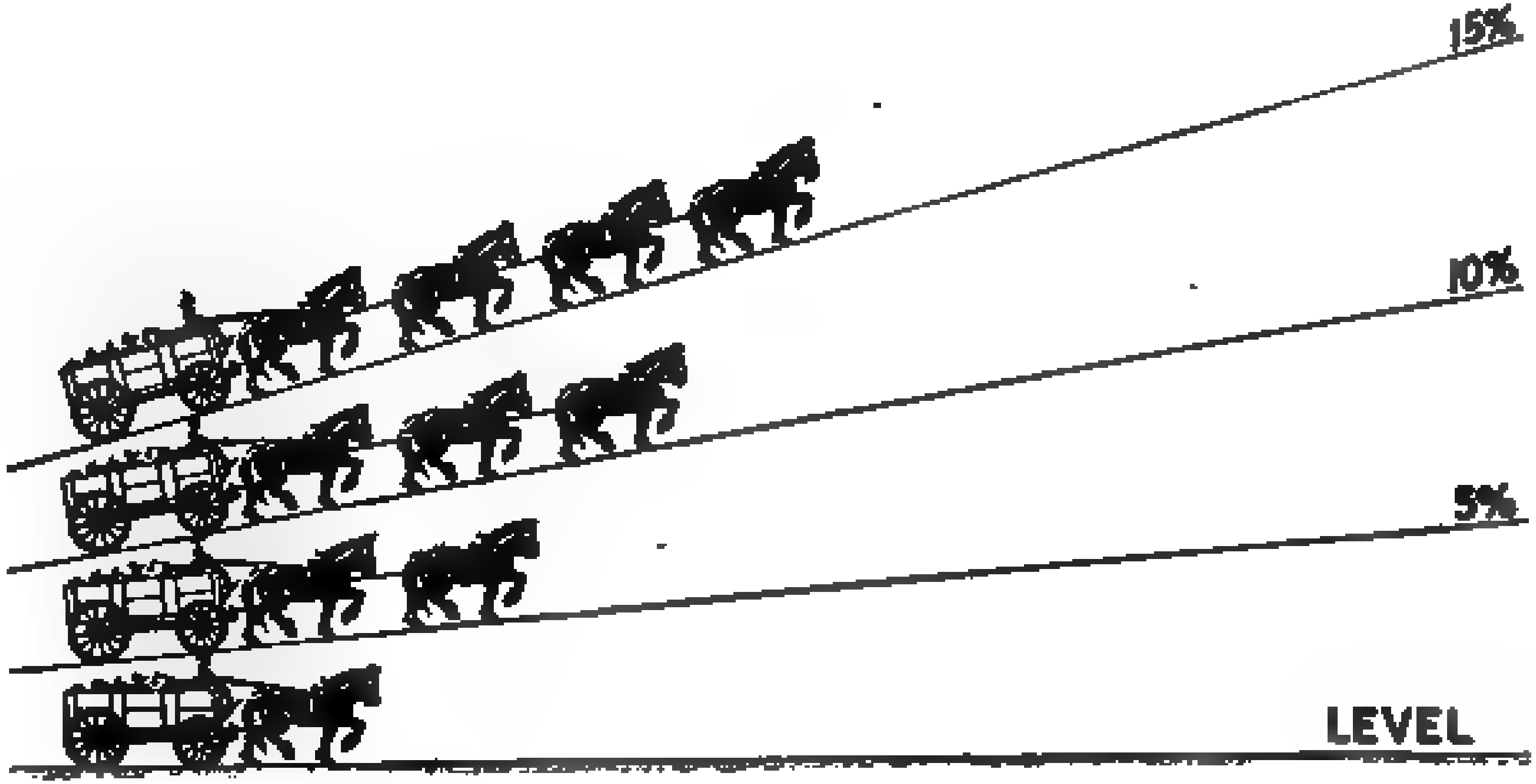
Fig. 1,716 is a sectional view of a single cylinder engine showing clutch, cross head pump, and heater.

The pump plunger is attached directly to the cross head and therefore operates in unison with the piston, making the same length stroke.

A two cylinder high pressure engine is shown in fig. 1,717.



Figs. 1,718c to 1,720c.—Geiser balanced piston valve. Fig. 1,718c. assembly; fig. 1,719c, bushing; fig. 1,720c, valve.



Figs 1 721 to 1,724.—Effect of

physical effort, two would be needed on a 5% grade, three on a 10%, and four on a 15% grade.

The advantage of this arrangement is the nearer uniform application or power, and the absence of dead centers owing to the cranks being placed at right angles.

Where better economy is desired, a compound engine is used; this type of engine is built in two, and four cylinder units. The former may be either tandem, or cross compound, while the latter

FIGS. 1,725 to 1,727. -Intercepting valve of the Reeves engine. When lever B, is pushed forward as shown in fig. 1,727, live steam is admitted to both cylinders, thus simpling the engine. In the position shown in fig. 1,725, the engine works compound. In the figures C, C, C, is a connecting bar to lever B. When the bar is pulled up into the position shown in fig. 1,725, the engine is compounded, thereby using its steam most economically, and also when compounding the reverse lever should be hooked up as close to the center as the work you are doing will permit; this will add further to the economy of the engine. C, C, C, connecting bar to lever B; E, steam pipe, F, throttle valve; I, operating rod to throttle; J, cross steam pipe; G, feed water heating chamber, H, exhaust chamber from high pressure cylinder to low pressure. The words simple and compound, with the arrows, show how to use the "intercepting" valve.

consists of two tandem compound engines connected to the same shaft with cranks at 90°. Most compound engines are so arranged that, in case of emergency, they can be converted into a simple double cylinder engine, both cylinders using

live steam. This will give a large increase in power for an extra heavy pull in ascending steep grades, or passing over heavy roads. The simpling device consists of an intercepting valve as shown in figs. 1,725 and 1,726.

It is located at A, and is of the same construction as the ordinary slide valve. When the lever B, is pushed forward into position, as shown in fig. 1,726, the live steam is admitted into both cylinders, that is, the engine has been converted into two simple high pressure engines, or *simpled*.

The engine is usually started with the lever in simple position; after the machine attains some speed, the lever is pushed back to compound position, thereby using steam most economically.

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FIG. 1,728.—Rumely double cylinder engine.

The eccentric (fig. 1,731) is provided with two parallel guide flanges that engage with corresponding flanges on a fixed disc (fig. 1,730) secured to the shaft by a set screw. To the hub of this fixed disc is pivoted an angle arm C, (fig. 1,729), one limb of which passes through an opening in the fixed disc,

FIGS. 1,729 to 1,731.—Best Mfg. Co.'s shifting eccentric reverse gear. The cut off may be adjusted by placing the eccentric in an intermediate position.

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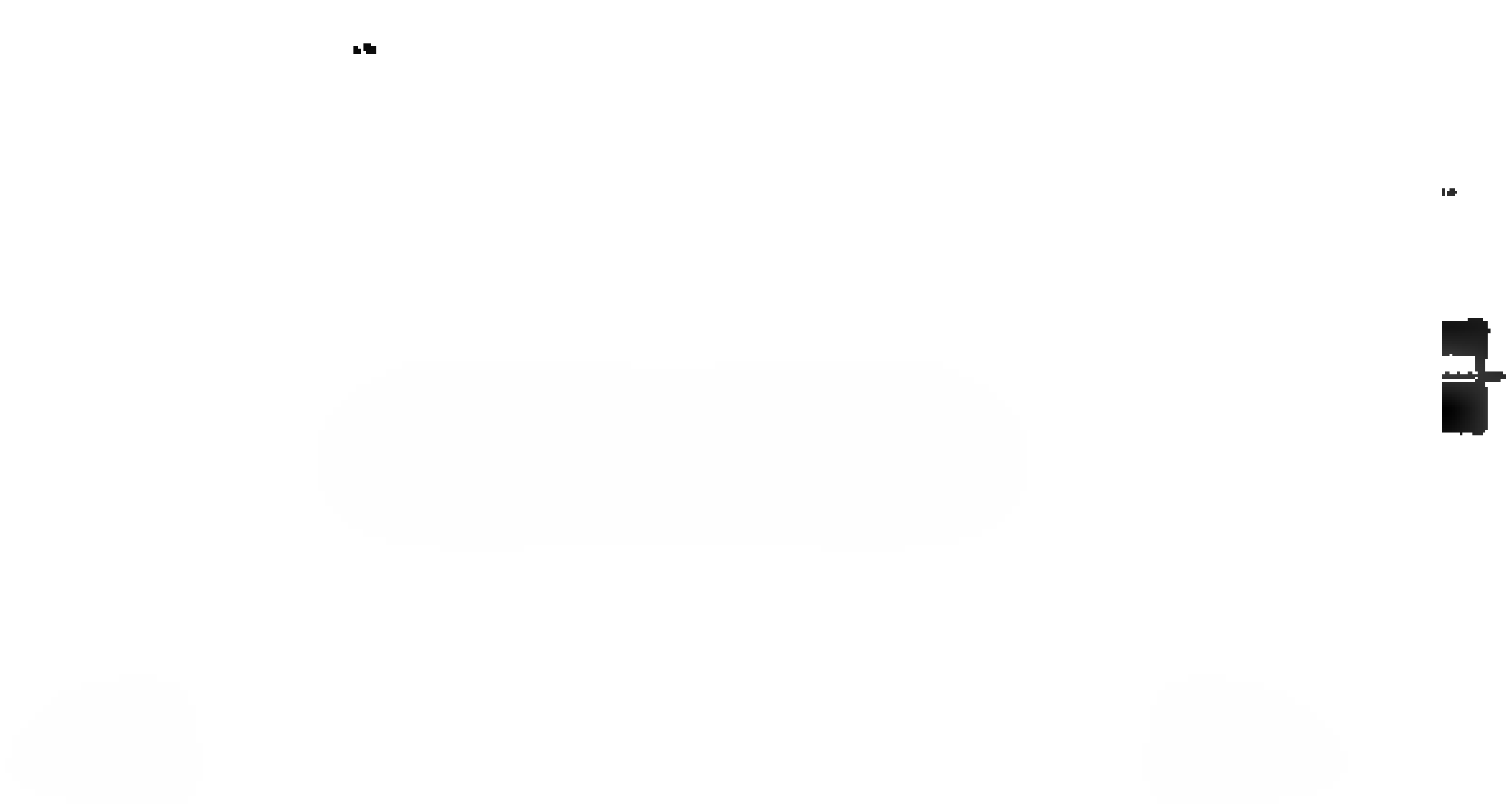


FIG. 1,733.—Geiser *shifting* eccentric. It is so constructed that at whatever point it is placed by throwing the reverse lever, it stands independent of any pressure from the collar until moved, and simply revolves as a part of the crank shaft.

FIGS. 1,734 to 1,738.—Marsh reverse gear as used on the Advance engine. It is composed of a three part box, one end of which is mounted on the main shaft, and at the other end is a small crank shaft. The crank of this shaft is connected directly to the valve, on the other end of the shaft is fastened a pinion which engages with another pinion on the main shaft. This gear is the equivalent of a *rotating* eccentric *which reverses by changing the angular advance, but since the throw is fixed it cannot be used for variable cut off*. To limit the travel of the box in reversing the engine, a stop plate is fastened to the frame of the engine, and the box in either of its two extreme positions rests against the screws in this plate. The lever for operating the reverse gear is mounted upon a quadrant at the rear of the boiler and within easy reach of the engineer. The quadrant is notched for forward and backward motion and neutral, and a spring catch is provided on the lever for holding it in position. The gear on the main shaft is in two pieces, and this allows the reverse gear to be easily taken off the shaft without dismantling the engine.

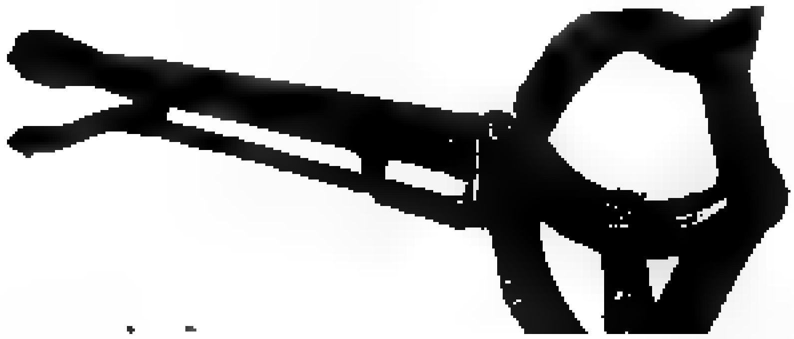


FIG. 1,739.—The Marshall radial valve gear as applied to the Reeves engine.

and extends within, and is pivotally connected within an opening to the shifting eccentric. The other arm of this lever is pivoted to a disc collar B, that is splined to move longitudinally on the shaft. A strap or arm F, from this disc collar connects with a slide rod G, parallel with the shaft.

This slide rod carries an adjustable collar with stud from which extends a connecting rod H, leading to the crank arm of a rock shaft. The latter may be operated in any suitable manner.

When the eccentric is to be shifted, the movement of the rock shaft causes the slide rod to shift the disc collar along the shaft, which in turn through the angle arm shifts the eccentric.

NOTE.—***How to set the Curtis valve gear.***—The pawls are first thrown out, to avoid damage in case the rods are not of proper length; the pawls are retained in the *out* position by slipping the ends of the pawl springs over the points of the pawls. The machine is then turned slowly by hand until the pawls on one set of valves are at their highest point of travel, then, with the valves wide open, the drive rods are adjusted so that there is one-thirty-second inch clearance at the point of the opening of the pawls when they are in. With drive rod jamb nuts set, the machine is turned slowly until the pawls are at their lowest point of travel. After closing the valves, each valve stem is adjusted to give one-thirty-second inch clearance at the point of closing of the pawls when they are *in*, securely locking the jamb nut as each valve is set. The operation is repeated on the other side of the machine, after which the governor rods should be adjusted. With turbine running and synchronizing spring in mid position, the rods are adjusted for normal speed at full load. The governor rods on the other side, (controlling valves 6 to 10) should be so adjusted that the speed change between the fifth and sixth valves will not be more than three or four revolutions.

FIG. 1,740.—Case ***geared feed pump.*** It is driven by a gear on the crank shaft meshing with a larger gear that carries the crank pin which operates the pump. This pump has sufficient capacity to supply fifty per cent more water than needed by the boiler in extreme conditions. The feed water on its way to the boiler passes through the steel shell heater. By means of a by pass valve, the water can be returned to the tank when not needed in the boiler. The feed can be regulated making it unnecessary to stop the pump.

,741—New Huber ***radial reverse gear.*** The slide is ʼided with a wood block.

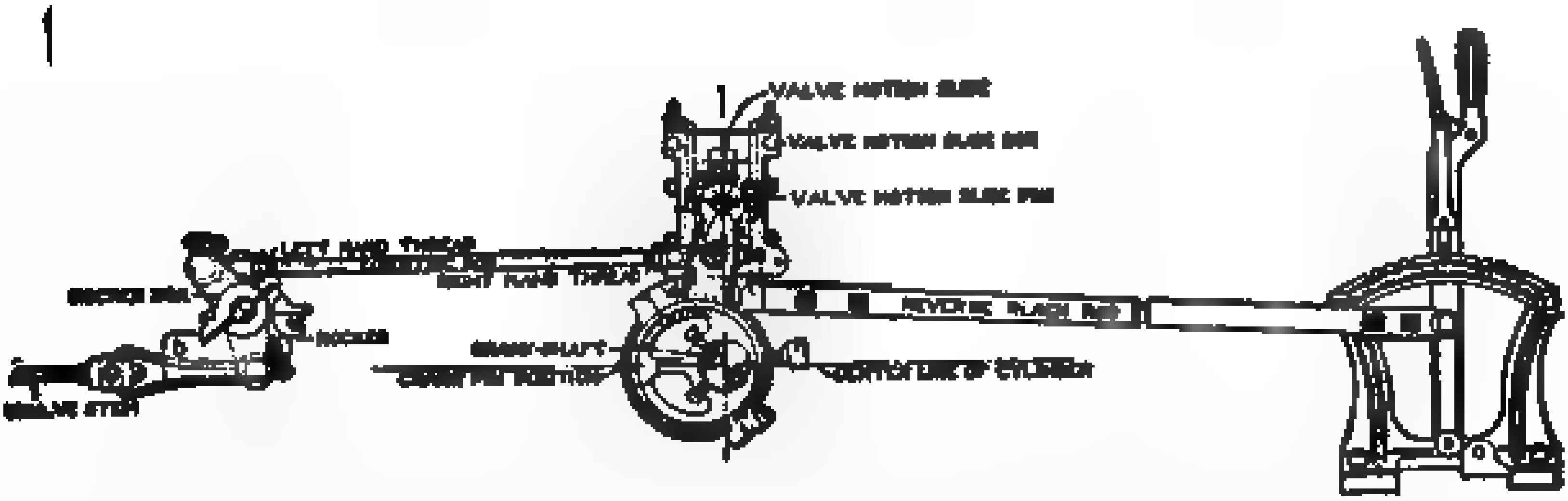


FIG. 1,742.—Port Huron ***radial reverse gear.*** The reverse lever quadrant has four notches at each end for hooking up and using steam on any of four different degrees of expansion running either forward or backward. A notch is also cut in center of quadrant, providing a safeguard against the engine starting by steam leaking past the throttle if the engineer, in stopping, set the lever in this center notch.

It will be understood that the disc carrying the shifting eccentric turns with shaft, the sliding collar disc being provided with a strap similar to an eccentric strap.

Gears of the so called radial class, are extensively used, the types usually employed being the Marshall, and the Bremme.

FIG. 1,743.—Huber traction engine. The boiler is of the return tubular type with superheater and internal cylindrical fire box as shown in fig. 1,751. The water tank is carried in front, and swings around so as to easily open the smoke box for repairs. Water is fed to the boiler by a cross head pump, and injector. The engine has one cylinder and is fitted with the Marshall valve gear, which is often erroneously called the Woolf gear. Dimensions of 25 horse power size: engine 10 × 12, 220 *r.p.m.*; speed reduction of transmission 21 1/3: 1; return tubular boiler: length 132, diam., 44, return tubes 22 — 2 × 105.

In fig. 1,739 is shown one construction of the former gear. This forms a simple and satisfactory reverse and one which is well adapted for traction engine service.

The link motion is too well known to require any extended description of its application here; a well designed link with

center suspension is shown in figs. 1,744 and 1,745, which illustrates also the location of the rocker and double reach rods.

Boiler.—The two classes of boiler in general use for traction engines are the direct tubular, or locomotive type, and the

FIG. 1,746.—Huber straw burning boiler. The straw chute is provided with a shutter opening inwardly, thus preventing admission of air when feeding straw. *In operation*, straw is at one end, and air th grates at the other end leg and baffle plate flame along the str causing rapid combust

FIG. 1,747.—Rumely link motion reverse. The gear is of the center hung locomotive type with link hangers provided with bronze bearings. The link, block, and connections are of steel. Notches are provided in the reverse so the engine can be hooked up to vary the cut off.

return tubular, a modification of the Scotch boiler; both have internal fire boxes, and, as built by the various builders, differ only in minor details. Boilers for traction engines are usually constructed so as to burn either straw, wood or coal.

For burning straw a change is generally necessary in the furnace such as putting an arch in the locomotive type, or an extension on the back end of the return tubular boiler so as to give the burning gases a longer distance to travel in passing through the boiler, and to provide more space for feeding the straw into the furnace. It also prevents the burning straw lodging against the ends of the tubes.

Fig. 1,746 shows the method of burning straw in locomotive boilers. A straw chute attached in place of the fire door, so that the straw may be fed continuously. A trap opening inwardly in the chute prevents the admittance of cold air. As the straw falls over the inner end of the grates, it comes in contact with the flame and is ignited, and burns while falling.

In the locomotive boiler, it should be noted that the firing door is low down, allowing the straw to be fed into the fire box immediately above the grate surface; also the fuel and draft enter from opposite ends, thus the flame, being guided by the arch, is led toward the incoming straw, resulting in rapid combustion.

An important requirement in boilers for traction engines is

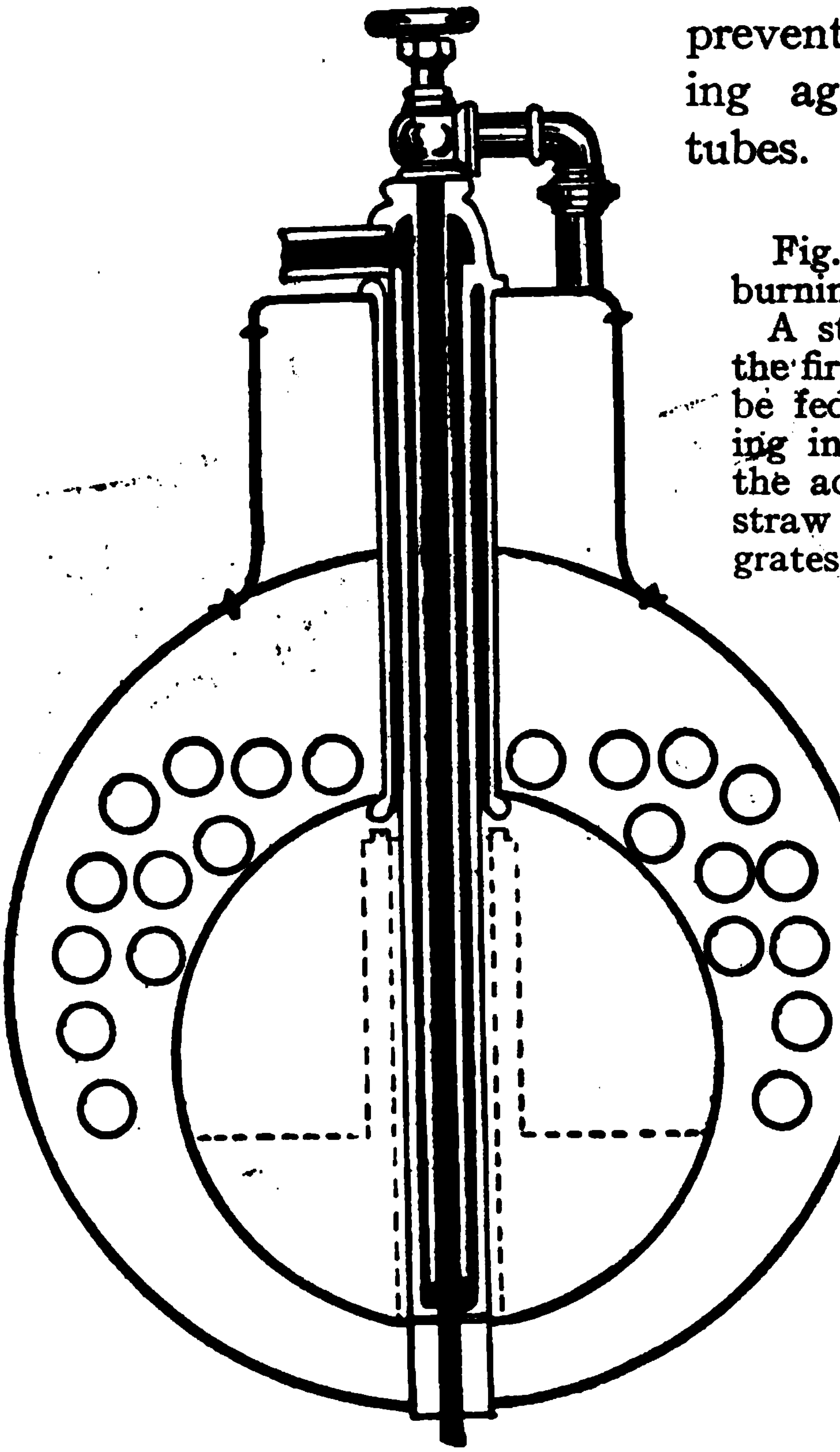


FIG. 1,751.—Huber boiler with superheater. Steam from the dome is taken in a small pipe down through the fire thence back in a larger pipe, thus superheating the steam.

some provision to prevent the crown sheet becoming bare of water when descending steep grades.

One method of doing this is shown in fig. 1,753. By means of a displacement chamber K, the water line does not reach below the crown sheet when the boiler is tilted as shown in the figure; the crown sheet is inclined so the end will at all times be covered with water. The dotted line Y, shows the water line as it would be without the displacement chamber.

Fig. 1,752.—Detail of Port Huron tube and sheet. The tube sheets are ½ inch thick with drilled holes. All tubes at the fire box end have copper ferrules.

Fig. 1,753.—Geiser locomotive type boiler. The figure shows the displacement device for maintaining the water level above the crown sheet in descending steep hills.

Some boilers are provided with superheaters to secure higher economy in the use of steam. A device of this kind is shown in fig. 1,751.

Steam is taken from the top of the dome in a small pipe down through the fire to the bottom of the fire flue, thence back in a larger pipe through the hottest part of the fire, thus superheating the steam before it is delivered to the engine.*

Friction Clutch.—The purpose of the friction clutch is to engage or disengage the engine shaft and the transmission

FIG. 1,757.—Wood friction clutch. The two friction shoes have wood frictions and adjustable friction rods which permit adjustment for wear.

gearing. It enables the engineer to move the machine very slowly, or apply all the power of the engine to the gearing at once when necessary to start a heavy load, or get out of a bad

*NOTE.—The saving in fuel due to superheating is approximately one per cent for each 11 degrees of superheat.

place or hole. Starting under these conditions being particularly difficult, the engine may be released from the transmission gearing and given high motion, then thrown into gear, the momentum gained giving power in starting largely in excess of the ordinary power of the engine. Fig. 1,758 illustrates the operation and construction of a clutch.

Keyed to the end of the engine shaft is a fly wheel, and concentric with this shaft and attached to the first transmission pinion is a double friction

FIG. 1,758.—Harrison six shoe friction clutch, with two speed change device. The construction is such that the friction spider can revolve with the fly wheel. When the engine is under the belt, simply throw the engine out of gear, and throw in the clutch, thus causing fly wheel and the friction spider to revolve together. The friction may be locked by loosening the set screws and dropping the two dogs, C C, which fit into corresponding recesses in rim of fly wheel. By doing this, all strain is taken off the shaft.

arm, which has iron shoes pivoted at its ends. These shoes have wood liners or frictions which engage with the rim of the fly wheel when pressed outward by the shoe rods, which are attached to a loose sleeve mounted upon a hub on the double friction arm.

There are two half rings which fit in a groove in the sleeve and to which is attached a fork which is connected with a lever mounted on the rear of the boiler for operating the clutch and which is within reach of the engineer.

The Transmission and Differential Gearing.—On account of the great resistance encountered by a traction engine in propelling itself along the road, or in ploughing, etc., it is necessary that the driving wheels turn much slower than the engine

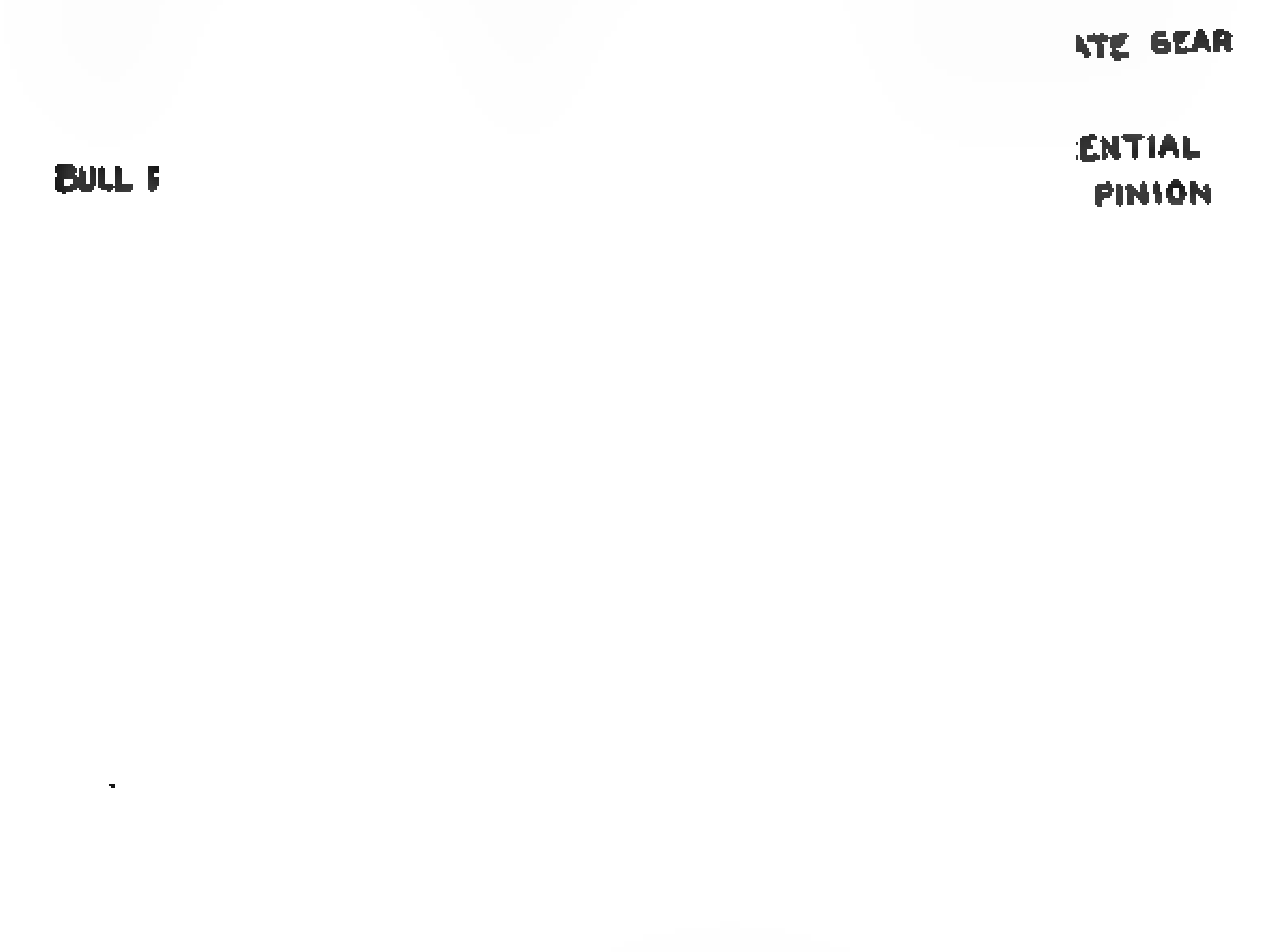


FIG. 1,759.—Wood transmission, and differential. Integral with the differential is a cushion device to prevent undue strain or shocks coming on the gearing in starting.

shaft. This reduction in speed is obtained by a series of gears known as the *transmission*, and shown in fig. 1,759.

A small spur *drive pinion* keyed to the engine shaft engages with the *intermediate gear* which being of larger diameter, gives the first reduction in motion. The intermediate gear, in turning, engages with the *differential*, which works on a counter

shaft. At each end of the counter-shaft is a bull pinion; these are in mesh with the *bull wheels*, thus transmitting the power to both drivers.

By observing the relative sizes of the gears it will be seen that the greatest reductions in speed are obtained by the first two and the last two gears. In some cases there is no reduction between the intermediate and the differential.

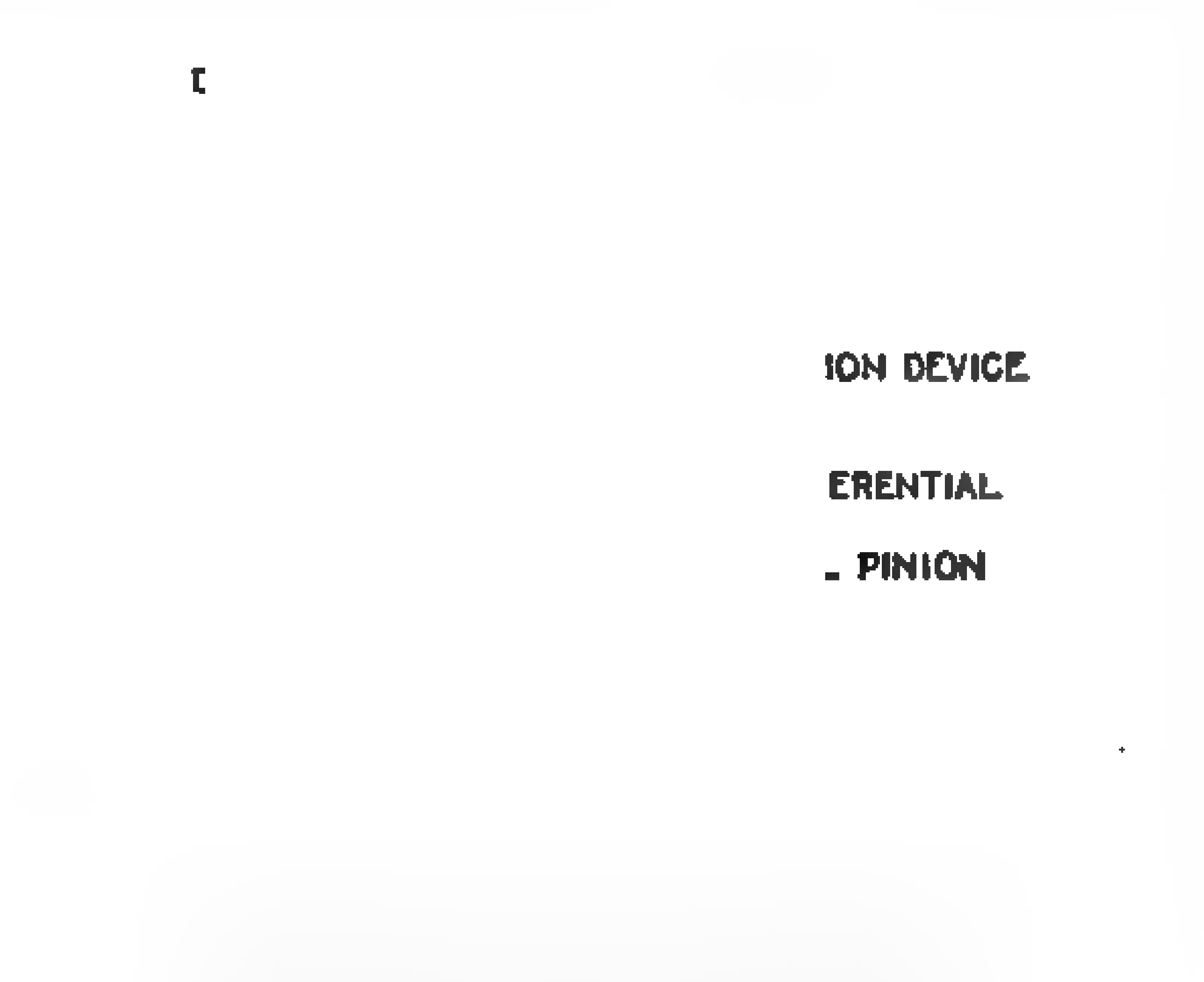
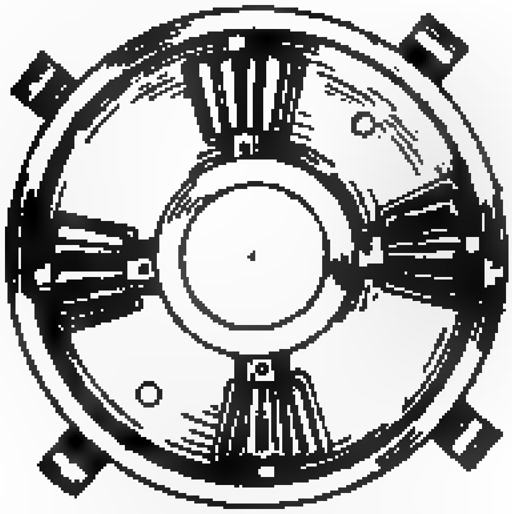
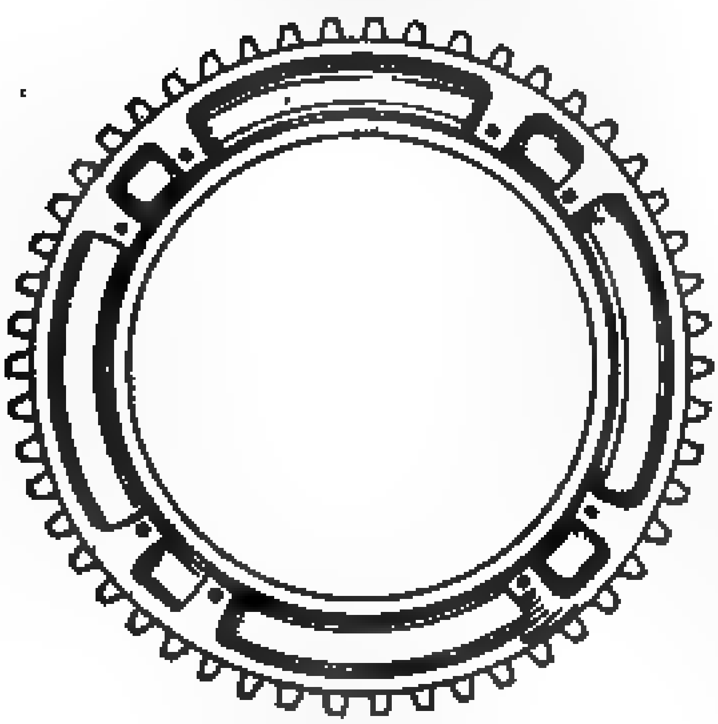


FIG. 1,760.—The Northwest friction clutch, transmission, and differential. The figure illustrates the large speed reduction between engine and drivers, which is necessary on account of the considerable tractive effort or thrust required to propel the engine and its load.

The special object of the differential is to transmit the power to both drivers, and at the same time allow them to turn independently of each other, that is, to allow one to revolve faster than the other when going around curves in the roadway.





Figs. 1,761 to 1 cushion device of the Wood traction engine. The four bevel pinions engage on one side with a fi countershaft, and on the other with a *loose bevel* cast with one of the bull pinions, and free to revolve on springs relieve the transmission of undue shocks in starting.

It must be evident that when the machine is making a turn this is necessary to prevent one wheel sliding since the path described by the outer wheel is an arc of a circle of larger diameter than that described by the inner wheel.

To secure independent turning of the drivers, the bull pinion shown at the left end of the countershaft is keyed to it, while the one at the other end is free to revolve on the shaft. This pinion is cast together with a bevel gear wheel called the *loose bevel*, and which is in mesh with four bevel pinions mounted on the differential, as shown in figs. 1,761 and 1,763.

On the opposite side of the differential (fig. 1,759) is a *fixed bevel* keyed to the countershaft, and which engages with the four bevel pinions. These bevels are of the same size.

In operation if traction power be applied to the differential, to move the machine straight forward, the four bevel pinions will remain motionless, and act simply as a kind of lock, or clutch, to secure uniform and continuous rotation of the drivers. If the machine be steered in a curved path, the outer driver will turn faster than the inner, causing the four bevel pinions to rotate on their axis. The inner driver, therefore, by the operation of

Figs. 1,764 to 1,766.—Port Huron differential gear costruction. The master gear A, is not keyed to the countershaft G, but turns it by force applied through the three bevel pinions B, and the respective bevel gears C and D. The journal for master gear A, is formed by the projecting hubs L, of the bevel gears over which it fits loosely. It turns on this journal *only* when one driver is revolving faster than the other. The bevel gear C, is keyed solid to the countershaft. Bull pinion E, is also keyed to countershaft. Set collar H, for taking end play out of countershaft, is held solid to the shaft with two set screws. Bevel gear D, is not keyed to countershaft but is an easy turning fit thereon. It is made in one casting with bull pinion F. It turns on the countershaft *only* when one driver is revolving faster then the other. At other times it turns *with* it. Countershaft nut J, holds bevel gears into mesh with bevel pinions. Bevel pinions B, do not revolve on their spindles K, when both drivers are revolving at the same rate. They do, however, when one driver is turning faster than the other. When one bevel gear (or drive wheel) is revolving faster than the other, the one that is turning the fastest gains on the master gear A, the same amount that the slower revolving bevel gear (or drive wheel) falls behind it. If one drive wheel be standing still and the other is slipping, the one that is slipping will be turning twice as fast as it would if both wheels were turning together at the same rate. The ***same tractive effort*** is being exerted at all times by both drive wheels—whether both are propelling, or both are slipping, or whether one is slipping and the other standing still. In the latter case, the force necessary to turn the wheel that is slipping is not sufficient to turn the one that is standing still.

the differential, is allowed to slow up, or remain stationary, as conditions require, while the outer one is urged forward.

On some engines the differential forms a part of the bull wheel; the construction of the differential movement varies in minor details, in some cases spur gears are used in place of bevel pinions.

FIG. 1,767.—View of Case spring mounting transmission gearing, frame for fuel bunkers, draw bars and connections.

An important part of the transmission is the *cushion device* which acts on the same principles as the spring draw bar of a railway car, being designed to protect the transmission from too violent and sudden strains, which might otherwise occur when the engine is started suddenly on a hard pull.

The operation of this device is shown in figs. 1,761 to 1,763. The outer toothed ring of the differential works on a disc upon which the bevel pinions are mounted.

The ring has a circular groove and radial projections or stops while the disc is provided with four projecting lugs which fit in the groove when the parts are assembled as in fig. 1,761.

Between the lugs and the stops are a series of stiff springs, hence, in the transmission of power the thrust of the ring is imparted to the disc and on to the drivers through the medium of these springs. The traction force is, therefore, gradually applied instead of abruptly, thus protecting the gears from severe strains and shocks.

FIG. 1,768.—Rumely engine. The cylinder is mounted at the forward end of the boiler and the shaft at the rear. The boiler is of the direct tubular locomotive type with round bottom fire box. Traction wheels of unusually large diameter are furnished. The engine is fitted with a shifting eccentric, reverse, friction clutch, cross head pump, and heater. There is an injector for auxiliary feed.

Steering gears.—The traction engine is guided by the front wheels which work on a pivoted axle the same as an ordinary vehicle. This axle is connected with a transverse shaft by two chains which are wound around the shaft right and left handed, and securely fastened. Hence, rotation of the shaft winds one chain and unwinds the other, thus turning the front axle. At the end of the shaft is a pinion which engages with a worm

gear on the end of the steering rod. There are several methods of operating the steering gear, as by:

1. Hand;
2. Friction from main engine;
3. Independent steam power.

Hand operation is quite common on medium, and small size machines, but when drawing a heavy load over rough roads it

FIG. 1,769.—Geiser steering gear. Two chains from the front axle are wound, in opposite directions, on a transverse steering shaft. This is operated through the worm gearing at the end by means of the steering wheel located at the end of the boiler.

requires considerable exertion, especially with large engines.

Fig. 1,769 shows the usual arrangement for hand steering. The transverse shaft is located under the boiler shell at the forward end of the fire box. Attached to the end of the steering rod is a hand wheel conveniently located for the engineer.

The second method of steering is illustrated in fig. 1,770, power being furnished by the engine.

A vertical shaft is a horizontal mitre gear arranged to engage alternately with two vertical bevel gears. These vertical gears are on a shaft by a chain of small gearing from the engine shaft. They are thrown in and out of gear by means of a shifting yoke, which is worked by a straight rod extending back to the right band side of the engineer and

Fig. 1,770.—Huber, friction steering gear. A vertical shaft leads from the worm gear to a horizontal shaft arranged so that the horizontal bevel at its end may engage alternately with the two vertical bevels on the horizontal shaft; the latter by a chain of gears is operated by the engine. The vertical gears are thrown in and out of gear by a shifting yoke with suitable connection, for operation by the engineer. The direction in which the front axle is turned in steering depends on which bevel is thrown into gear.

having a lever at the end. By moving it forward or backward, the engine is guided to the right or left, as desired.

An independent steam steering gear is shown in fig. 1,771, this is a desirable method for large engines.

It consists of a double cylinder engine geared to the transverse steering shaft. The valve gear is of the single eccentric type; there is a bell crank connection to a collar and sleeve on the shaft running to the steering wheel in the cab.

The engineer, in steering, turns the wheel to the right or left according to the direction he wishes to go. The action of the valve gear is such that the steering engine operates while the wheel is turning and stops when the motion of the wheel ceases, the steering gear remaining in that position until a change is desired.

FIG. 1,771.—Avery independent steam steering gear. The steering axle is connected by spur gears to a two cylinder engine having cranks at 90 degrees. The action of the valve gear is such that the engine turns the steering apparatus to the position corresponding to that of the steering wheel, and retains it there until the position of the latter is again changed.

Feed Pumps and Heaters.—The most usual, and in fact the best method of supplying the boiler with water is by a pump driven direct from the cross head of the engine, as shown in fig. 1,772. This type of pump is single acting, and with the plunger properly proportioned for the work, the operation is ideal.

On the discharge pipe is a *by pass* valve leading to the tank, so that the quantity of water being fed to the boiler can be regulated, or all returned to the tank as required. Hence, water may be fed continuously to the boiler, and in the same amount as used.

A cross head pump will operate on from one-half to one-third the amount of steam required by a small direct acting steam pump;* while it has the advantage of economy, it can operate only while the engine is running.

An independent fly wheel pump is sometimes used for feeding the boiler, as it works with practically the same economy as a cross head pump, and has the desirable feature of being able to operate when the engine is not running.

FIG. 1,772.—Huber cross head pump. Some advantages of this type of pump are that it supplies the boiler with water as used, and being worked by the engine is economical in operation. Much heat is saved by using a pump and heater instead of an injector as the feed water is heated by exhaust, instead of live steam.

Figs. 957 and 958 show a well designed pump for this purpose. It is arranged to be operated either by hand or steam, and since the speed may be varied, the water supply may be regulated without a by pass.

*NOTE.—A well designed single cylinder traction engine will consume from 50 to 60 lbs. of steam per hour per horse power, while a small direct acting boiler feed pump will require from 120 to 200 lbs. on account of working without expansion, slow speed, leakage, slip and inefficient valve gear.

Most traction engines are provided, as they should be, with two means of feeding the boiler so that if one break down, the other is available. The boiler should be fed regularly. The reason for using a pump in preference to an injector is because considerable fuel is saved by heating the feed water with the exhaust steam.*

FIG. 1,773 and 1,774.—Northwest fly wheel boiler feed pump. This method of feeding the boiler is practically as economical as the cross head pump, and has the advantages of direct instead of by pass regulation, slower speed and of independent working, permitting the boiler to be fed when the main engine is not running.

*NOTE.—The saving due to heating the feed water is approximately one per cent for each increase of 11 degrees in the temperature of the feed water. A well designed and properly proportioned heater will raise the temperature of the feed water within a few degrees of the temperature of the exhaust. Thus, if the water enter the heater at, say, 60, and be heated to 210°, the saving is $(210 - 60) \div 11 = 13.6$ per cent.

The discharge from the pump is piped to a heater usually located on the side of the boiler shell, from which it is delivered to the boiler. One type of heater for traction engines is shown in fig. 1,775.

It consists of a cast iron shell containing a number of tubes which are expanded in the heads at the ends. In each tube is placed a pipe extending

FIG. 1,775.—Case feed water heater. A number of tubes are expanded into the two heads of the shell. The feed water passes through thin annular spaces, formed by placing inside the tube, pipes of somewhat smaller diameter; it is thus brought into intimate contact with the tubes, thus extracting considerable heat from the exhaust steam as it passes over the tubes. The saving due to heating the feed water is approximately, one per cent for each 11 degrees elevation in temperature of the feed water.

FIG. 1,776.—Huber feed water heater consisting of a coil "made of pipe and return bends placed inside a tight vessel or receiver."

full length, and which forms an annular space, about one-eighth inch thick, through which the feed water passes; with this arrangement, considerable heat is absorbed from the exhaust steam which enters the shell at one end, flows around the tubes, and passes out at the other.

Drain cocks are provided at lower part of the shell so that it may be properly drained when not in use.

How to Operate a Traction Engine.—It is usually best to carry the water a little higher while on the road than when the engine is at work, especially in hilly country. There is

FIG. 1,777.—Huber exhaust relief. Its purpose is to give a sharp exhaust to increase the draught when burning poor fuel, or when it is desirable to raise steam quickly. The main nozzle is closed nearly to a point to make the exhaust sharp. By opening a globe valve, a part of the exhaust passes through the relief chamber. In this way the operator has control of the draught and can prevent the throwing of sparks.

more or less fluctuation of the water on an uneven roadway, and unless the boiler contains some special provision for maintaining the water level (as, for instance, some device such as

FIG. 1,778.—Throttling governor as used on the Huber engine. It is placed horizontally on the engine to eliminate bevel gears.

shown in fig. 1,753), the tube ends or crown sheet may become bare when on a hill. Care, however, should be taken not to carry the water high enough to cause priming.

The steam pressure should be maintained near the blow off point. The valve gear, if of the expansion type, should be well hooked up, so that steam may be used with as much expansion as the running conditions permit, to secure the economy resulting from a short cut off.

Before descending a hill it is well to put fresh coal on the fire, and close the damper.

FIG. 1,781.—Rear view of Advance-Rumely rear mounted engine bed.

Since very little steam will be required this will prevent a loss of water by blowing off, and in case the crown sheet become exposed by the receding water it will be somewhat protected from the intense heat of the lower layers of the fire.

When descending the hill the speed of the engine should not be allowed to increase much above its ordinary speed. If the engine run too fast on a reduced throttle, the latter should be further reduced and the brake applied if necessary. In case the engine be not provided with a brake, it will be necessary to shut off steam, and reverse; this converts the engine into a pump, and the air, or rather smoke and cinders in the stack are carried through the heater and cylinder, being compressed therein, and forced into the steam chest. While acting as a resistance to check the speed, it is very bad for the cylinder and valve, and should never be done if it can be avoided.

As soon as the bottom of the hill is reached, the damper should be opened and the fire allowed to brighten.

FIG. 1,782.—Port Huron spark arrester, as used on straw burning outfits. The bottom sleeve is tapered and fits into the top end of regular smoke stack. The body sides and cap are made of heavy wire netting. A rod extending from cap to base of stack, where it is held by a bracket and tail screw, makes it easy for the operator to raise the cap when firing up, also to jar the soot out of the screen.

When approaching a hill, a good supply of water should be fed to the boiler, because in making the ascent it may be necessary to shut off the feed to maintain the steam pressure, if the hill be very steep.

Under ordinary running conditions, the feed should be continuous and at the rate in which it is used, rather than in spasmodic doses.

It requires skill to successfully take a very steep hill. In beginning the ascent, the steam pressure should be near the blowing off point, the engine well hooked up, and running slowly in order not to use the steam too

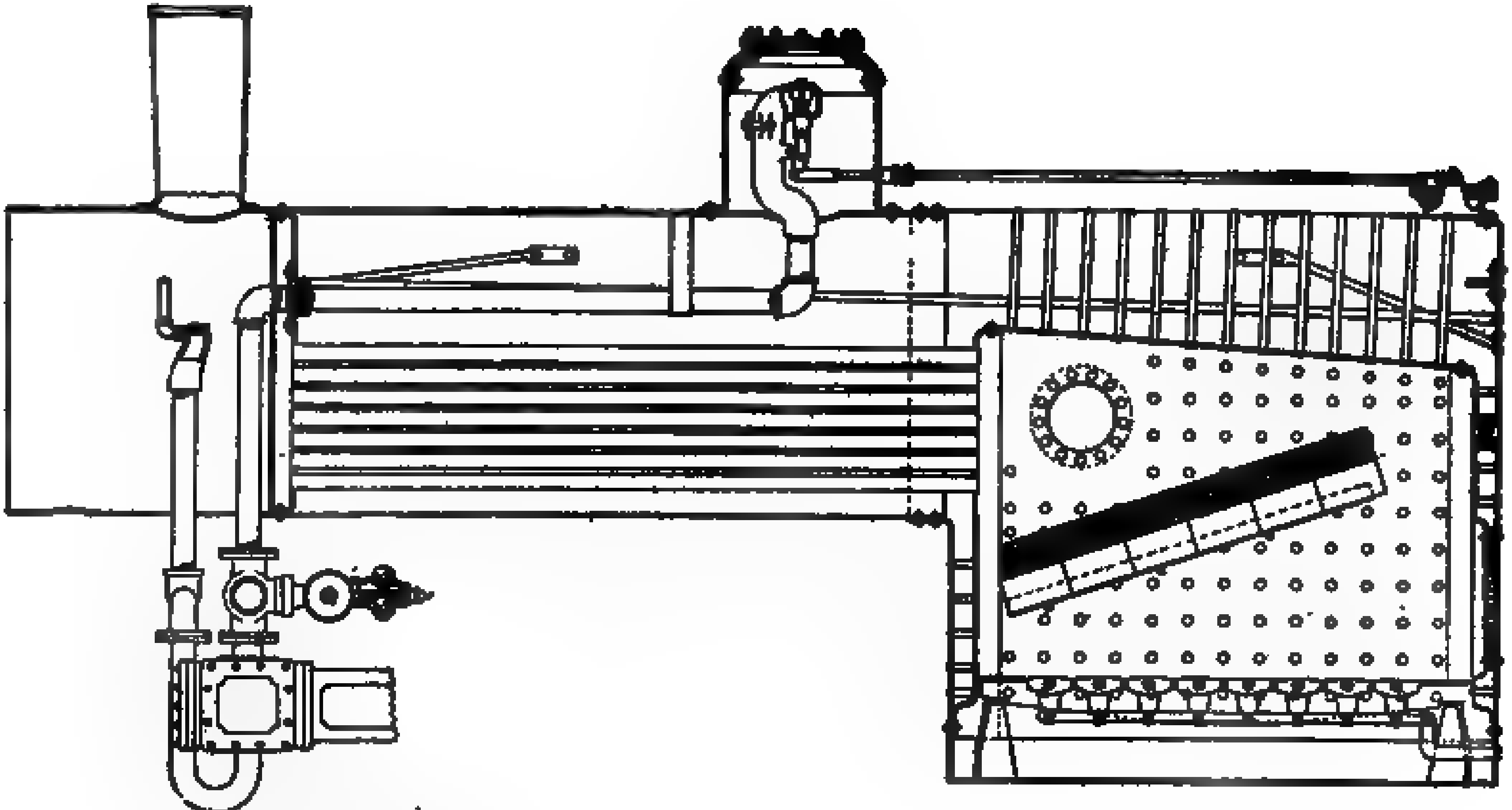


plate.

quickly at the bottom of the hill. As the grade increases, the speed should be maintained by first gradually opening the throttle; when the throttle has been fully opened, the cut off is lengthened by degrees as
r is required.

It is well to remember that the steeper the hill, the slower should the ascent be, in order not to overtax the boiler and cause the steam pressure to drop. No firing should be done while going up a hill, because the first effect of adding fuel is to check the formation of steam.

If the hill be a long one and it be necessary to fire while ascending it, the firing should be done near the bottom.

On ascending a short hill the water supply may be shut off; on a long hill the feed may be reduced, and shut off near the top if necessary.

In steering an engine, the engineer should keep his eye on the front wheels and note their position. As usually constructed, the machine will turn in the same direction in which the top of the steering wheel is turned.

The steering wheel may be turned more easily when the engine is in motion than when standing still. The steering chains should be moderately tight, though not tight enough to cause undue friction. If too slack, it will be somewhat difficult to guide the engine.

FIG. 1,784.—Rumely concaves and concave adjusting device. By far the greater part of separation is done at the cylinder and immediately behind it, before the straw reaches the rack. The concaves have diagonal ribs, so arranged that in conjunction with the teeth, maximum separation is accomplished. Much grain is hurled by the motion of the cylinder through the grates and the inclined slattered carrier behind them. The straw is conveyed up this carrier, being agitated meanwhile by a large winged beater which revolves directly above it. The concaves can be adjusted to any condition of grain. This is done from the outside of the machine by means of a worm gear device seen at the right of the cut.

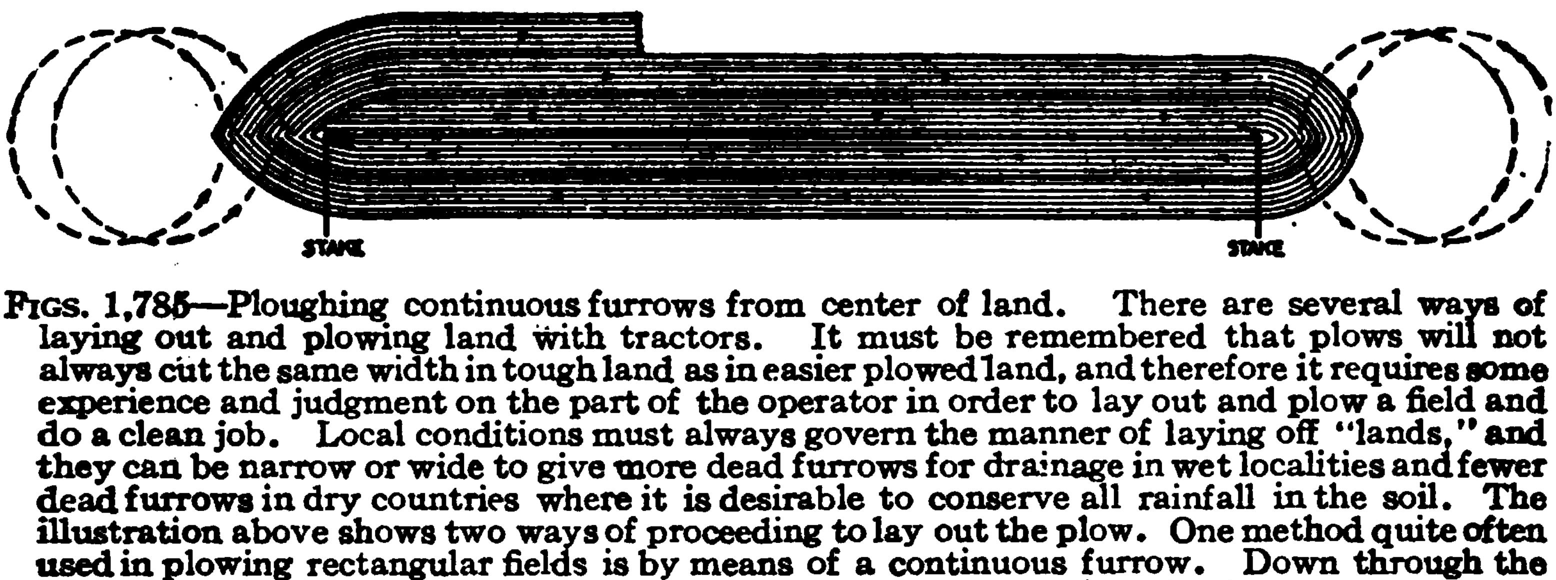
Mud holes should be carefully avoided as the wheels will have a tendency to slip.

If the wheels turn without propelling the engine, the calks on the wheels will dig holes, making it more difficult to get out. The engineer, then, should stop the engine when the wheels begin to slip, and put something under the wheels, such as straw, rails brush, or any kind of rubbish, that will

enable the wheels to get a grip upon the earth. Sometimes a log chain is fastened to some object ahead of the engine and one end thrown under the wheel.

In extracting the engine from a hole, it should be run slowly as it is less liable to lose its footing.

Before crossing a bridge, the engineer should examine it, to see if it be strong enough to carry the weight of the engine and load. An engine may be run across a weak bridge safely by



FIGS. 1,785—Ploughing continuous furrows from center of land. There are several ways of laying out and plowing land with tractors. It must be remembered that plows will not always cut the same width in tough land as in easier plowed land, and therefore it requires some experience and judgment on the part of the operator in order to lay out and plow a field and do a clean job. Local conditions must always govern the manner of laying off "lands," and they can be narrow or wide to give more dead furrows for drainage in wet localities and fewer dead furrows in dry countries where it is desirable to conserve all rainfall in the soil. The illustration above shows two ways of proceeding to lay out the plow. One method quite often used in plowing rectangular fields is by means of a continuous furrow. Down through the center of the field, stakes are set 10 or 15 steps closer to the ends than the sides, to allow for the narrowing of the furrow in turning the ends. Plows are started at the left end and pulled to the right, in the direction of the arrows. When the opposite stake is reached the plows are lifted and the tractor turned around as indicated by dotted lines, and swung in on a curve so as to round up the ends. It is necessary at the outset to make this loop at the ends on account of the short turn on the return furrow. When the land is wide enough the plows are left in the ground for a continuous furrow around the entire land. When the field is finished nothing will remain except small corners which can then be finished. From the standpoint of saving time this is the most approved method, because the plows are working continuously. Curve plowing, however, due to an uneven distribution of the load, is hard on the tractor gears.

placing thick planks lengthwise of the road so the weight will be distributed over a larger surface.

In starting an engine on the road, it should be first brought to speed, and then the friction clutch thrown in gradually until the gearing is in motion, when the friction may be set tight.

In reversing while in motion the throttle is first closed, the reverse lever thrown over, and the throttle gradually opened again. It is done in this way so as not to reverse too quickly which would bring severe strains on the transmission.

An important maneuver in handling a traction engine is "setting the engine."

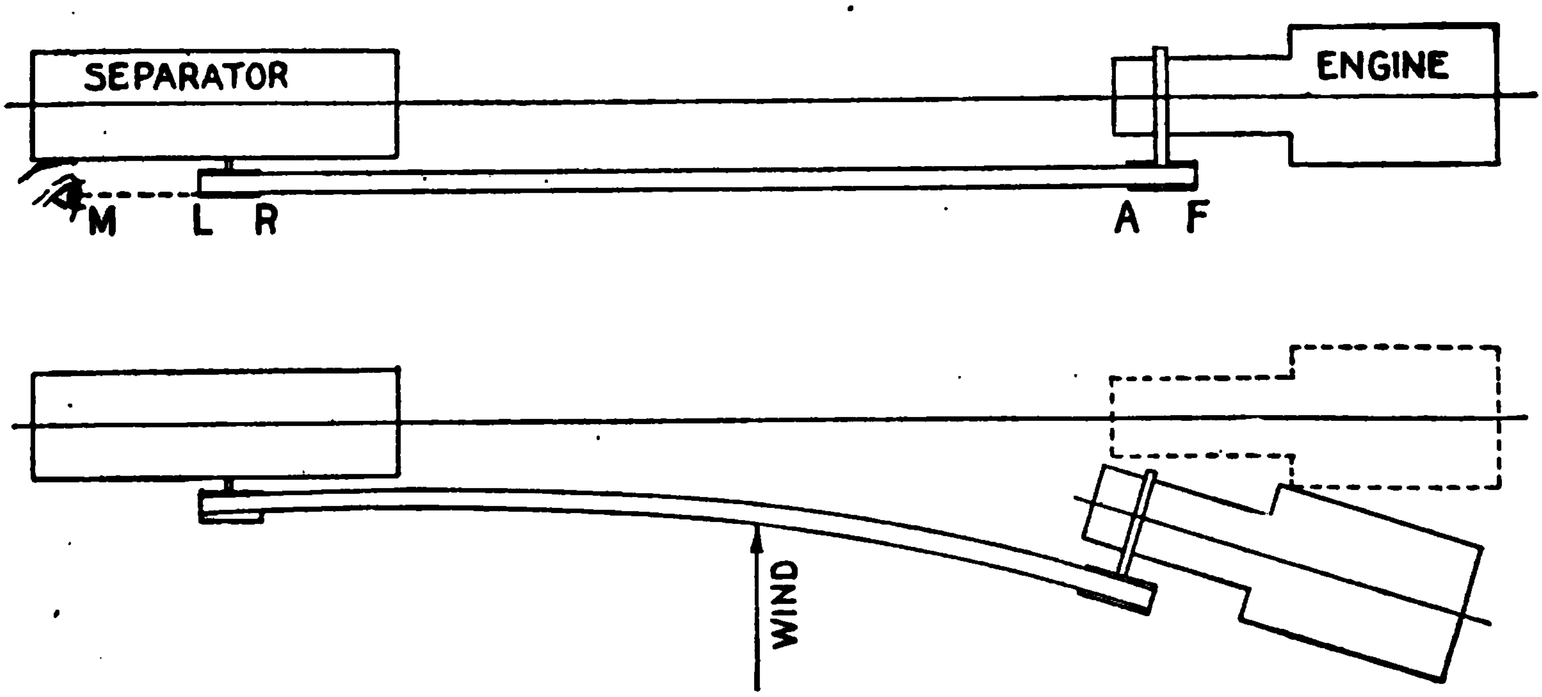
FIG. 1,787.—Case tractor at work. The engine is of the two cylinder opposed type, size 8 × 9 magneto ignition; force feed lubrication; thermo siphon circulation. The radiator fan has a friction drive with the engine fly wheel. Two speed transmission.

Nearly all threshing outfits are arranged so that the engine pulls the water wagon, fuel tender, and separator while on the road. The separator is pulled in to the stacks, the engine uncoupled, and then turned and placed in position. The latter operation is usually done by running ahead to the right or left, until at right angles with the separator. The engine is then reversed and backed in the other direction until about the required position. It should be backed a little farther than required if not backed into exact line, and then brought up in line with the separator pulley.

FIGS. 1,708 to

When the wind does not blow from either side, the engine should be in exact line with the separator; it may be determined if the separator be in line by sighting with the eye, as shown in fig. 1,792. The engine should be manœuvred, until the points L, R, A, F, are in line, L, representing the point of sight.

If there be a strong side wind, it is well to set the engine a little to one side toward the wind with the fly wheel turned slightly away. The wind will tend to carry the belt with it,



FIGS. 1,792 and 1,793.—Setting the engine. In fig. 1,792, the separator and engine are placed in line by manœuvring until points L, R, A, F, of the pulleys are in line, as sighted by eye at M. In fig. 1,793, when there is a strong side wind, the engine is set a little to windward with its fly wheel turned slightly away from the wind, to allow for the action of the latter on the belt.

and the engine should be placed so the belt will run straight with the fly wheel and separator pulley as in fig. 1,793.

The fly wheel on nearly all traction engines turns ahead while threshing, and backwards while travelling on the road. If the friction clutch be thrown in slightly while threshing, the belt may be tightened at any time. A good sized block, or a jack screw should be carried to block the drive wheels when threshing.

CHAPTER 35

ROAD ROLLERS

In the construction and repair of roadways, a modified form of traction engine is used for the various operations, such as: plowing up old roads; "spiking" or loosening the surface of worn out macadam roads; rolling or compressing the road to make it hard; consolidating earth fills and embankments, etc. There are two classes of road roller, adapted to special conditions in road construction:

1. The tandem, or two wheel roller;
2. The three wheel roller.

The Tandem Roller.—This type of roller was first designed for use in rolling newly laid asphalt, or brick pavement. In addition to this service the tandem roller is used to advantage in finishing or "puddling" a macadam road, where it is employed as an auxiliary to the three wheel roller; also in building roads and driveways for light traffic, in rolling lawns, or golf courses. The center of gravity being low it may be operated on very rough ground without danger of upsetting.

For rolling asphalt, the rolls are turned in a lathe and set true in the bearings, thus, the pavement may be rolled without waves or depressions. The rapid movement of the rollers adopts itself well to this class of work. This feature is of ad-advantage in finishing a macadam road where the tandem

roller can follow directly behind the sprinkler and get the advantage of the entire amount of water. Fig. 1,794 shows a typical tandem roller.

The main frame is constructed of steel channels, curved at the front end to form a "goose neck" for the king pin bearing of the forward roller. The horizontal part of the frame lies quite low being under hung from the axle of the rear roller.

The main bearings, at either side of the roller carry the greater part of the weight, and consist of heavy castings bolted to the frame. In the

FIG. 1,794.—The Erie tandem road roller. It is fitted with a two cylinder high pressure engine with bevel gear transmission power steering device, and sleeve reverse. Rollers of this type are usually built in sizes ranging from 2½ to 10 tons.

design shown in the figure, the cap is on the lower side, bringing the weight on the solid casting instead of on the studs which hold the cap.

The boiler is of the ordinary vertical shell type, the usual working pressure for this service being 150 lbs. There are, as a rule, two injectors for boiler feed, and in some cases an injector, and cross head pump; the latter combination is preferable, if a feed water heater be provided.

The engine consists of two high pressure slide valve cylinders with cranks at 90 degrees to avoid dead centers. The reverse is made by a spiral slotted sleeve, cast solid with the eccentrics and free to revolve on the crank shaft.

The saddle over the front roller is made of extra heavy flanged steel, and swivels at the center of a steel universal joint, in which the vertical pivot or king pin (corresponding to the king bolt in a carriage) allows the axle to turn, for the purpose of guiding the machine, and also permits the front axle to assume any angle with the rear axle. This allows the roller to adapt itself to uneven surfaces without twist or strain on the frame, or possibility of injuries to the engine.

The king pin is a steel forging of large diameter; its bearings or journal box is made in two parts to permit adjustment.

The transmission, as shown, consists of a beveled pinion at the end of the crank shaft which engages with beveled teeth on the side of the rear roller.

FIG. 1,795.—Iroquois 2½-ton tandem roller with attachment for rolling asphalt.

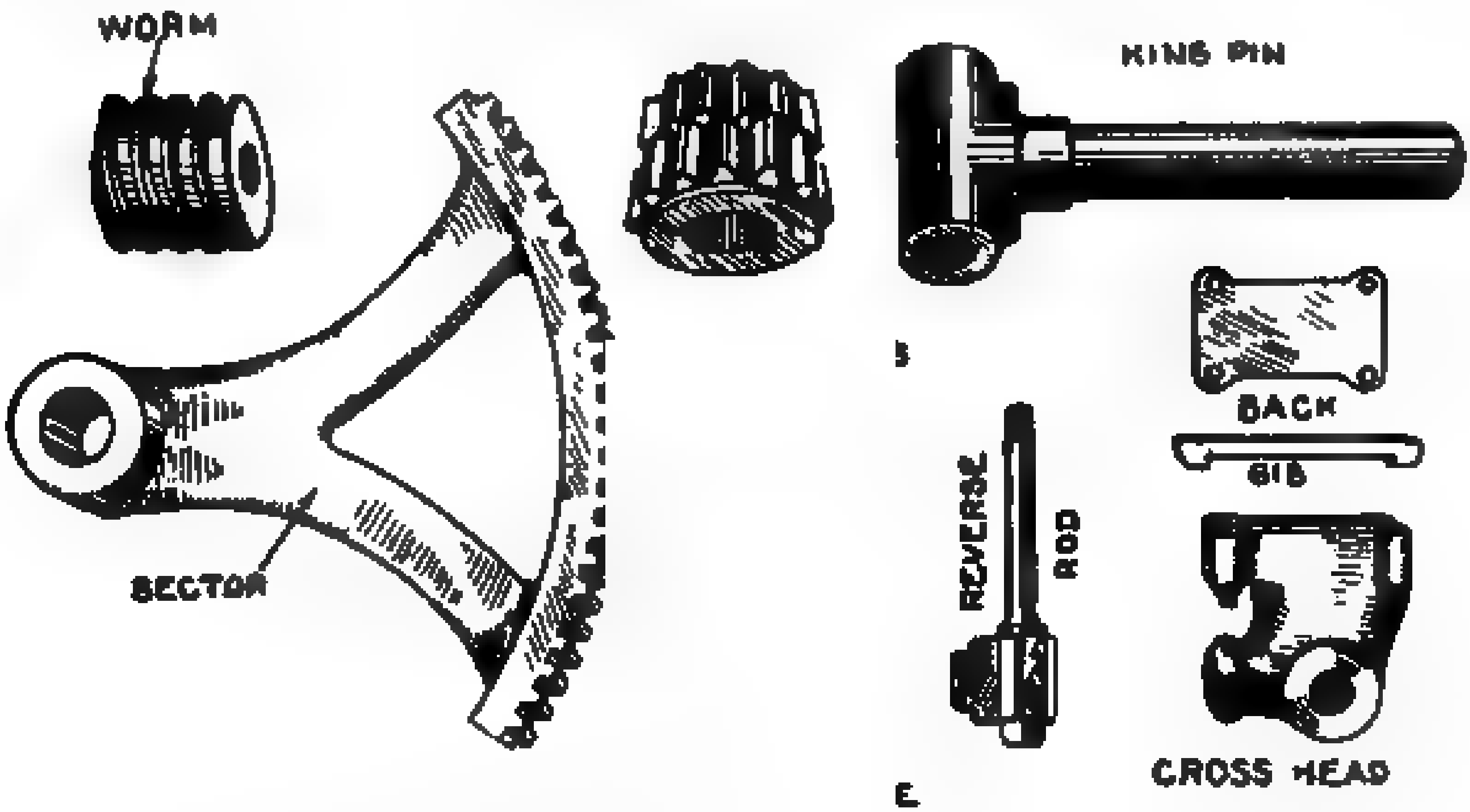
On account of the conditions of operation a considerable speed reduction is here made between the crank shaft and rear roller. Both pinion and gear have two step teeth arranged for alternate contact between the two sets to give smooth working.

NOTE.—Iroquois 5-ton tandem, standard roller. *General dimensions:* Length over all, 13 ft. 8¼ in., width over all 4 ft.; height over all 8 ft. 9 in.; height with smoke stack removed: 6 ft. 8 in.; center to center of rolls: 8 ft. 4 in.; diameter of driving roll: 4 ft.; face of driving roll: 3 ft.; diameter of steering roll: 2 ft. 6 in.; face of steering roll: 3 ft. 1⅛ in.; diameter of front axle: 2⅝ in.; diameter of rear axle: 3½ in.; diameter of engine shaft: 2¾ in.; size of crank pin: 2 in. × 2¼ in.; size of wrist pin in cross head: 1 in. × 4⅛ in.; diameter of steam pipe: 1¼ in.; diameter of exhaust pipe: 1¼ in.; diameter of boiler shell: 2 ft. 4 in.; height of boiler: 4 ft. 6 in.; number of tubes: 84; size of tubes: 1½ in.; length of tubes: 2 ft. 10¾ in.; sq. ft. heating surface. 101.36; sq. ft. grate area: 3.14; ratio of heating surface to grate area: 32.2 to 1; working steam pressure: 125 lbs.; size of engine (two): 6 in. × 6 in.; normal speed of engine: 124 R. P. M.; ratio of gearing. 5.88 to 1; coal capacity: 228 lbs.; tank capacity: 116 gals.; compression per inch of width, rear rolls: 212; compression per inch of width, front rolls: 92; average speed in miles per hour: 3; weights: actual manufactured weight: 9,300 lbs.; weight with coal and water: 11,020 lbs.

FIG. 1,798.—The Harrisburg, three wheel road roller. The boiler is of the locomotive type, and the engine has two high pressure cylinders. The construction of the three wheel roller is practically the same as the traction engine, except that there is a roller instead of two forward wheels. The usual range of sizes is from 10 to 15 tons.

The steering gear consists of a steel segment forming a worm gear keyed to the king pin, the worm shaft being fitted with two hand wheels. A friction power steering device is provided on the larger size rollers as shown. A sprocket on the worm shaft is connected by chain drive to a pinion which meshes with two bevel wheels on the engine shaft. These bevels turn loosely on the shaft, but either may be thrown into frictional engagement with the shaft by the lever and friction clutch, the direction of steering depending on which bevel is in operation.

The water supply is carried in a metal tank mounted on the rear end of the main frame, and arching slightly on the inner side to conform to the arc of the roller. The capacity of tanks on tandem rollers is usually sufficient for five or six hours' continuous operation.



FIGS. 1,797 to 1,805.—Some important parts of a tandem roller: 1, worm for steering gear; 2, sector for steering gear; 3, reverse sleeve and eccentrics; 4, reverse sleeve rod; 5, King pin; 6, cross head; 7, gib; 8, cross head back.

A coal bunker is placed above the water tank, extending forward to the boiler. On some rollers there is a bunker below the foot plate. The size of bunker is usually large enough to carry a ten hours' supply of fuel.

The Three Wheel Roller.—This class of roller is adapted to building macadam, telford, gravel, shale, or dirt roads; it is useful in pulling a plow, spiking, and for driving a stone crusher or as a road locomotive in hauling a train of wagons. Fig. 1,796 illustrates the three wheel roller which is composed of three principal members: engine, boiler. and rollers. The construction

of these machines is practically the same as the traction engine with the exception that the wheels are replaced by rollers.

The front roller is in two or more sections on a single axle and is pivoted in the same manner as on the tandem roller for steering. The several sections enable the roller to turn without twisting or rooting the road material. Approximately two-thirds of the weight of the machine rests on the driving or rear rollers, which are therefore the chief agent for compressing the

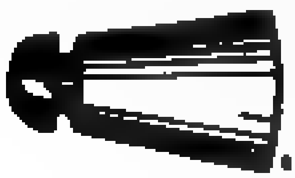
FIG. 1,806.—Detail of Case roller showing transmission gearing. Power is applied to both drivers through the large periphery of the ball gears direct to the rims; the pull being applied through the tangential bars which connect the large gear to the rim as shown.

road material. These wheels are of large diameter so that obstructions may be easily surmounted without jar, and that the fresh material will not be crowded in front of them. When the machine is travelling in a straight direction the driving rollers overlap the track of the steering rollers by about six inches.

The driving rollers may be fitted with a set of spikes as shown in fig. 1,807, for loosening the surface of the old roads; a set of plugs is provided for filling the holes when the spikes are not in use.

Fig. 1,807.—Rear roller, with spikes inserted for "spiking" or loosening the surface of worn out macadam roads. The spikes are easily removed, and when not in use plugs are inserted to fill the holes in the tire.

Fig. 1,808.—Russell compound steam road locomotive with special tanks, engineer's cab and canopy. Curtains can be attached to drop over sides and enclose the engine. Cable drum attached. ***Details of 50 rated horse power size:*** Working pressure, 160 lbs.; *i.h.p.*, 120; cylinders, 8 and 14×12; *r.p.m.*, 300; speed on road, 2 to 4 *m.p.h.*; fly wheel, 42×14. ***Boiler:*** Diam., 34; tubes 45—2½×72; fire box, 30×52; grate area, 10.8 sq. ft.; heating surface, 250 sq. ft. ***Tanks:*** Water, 500 gals.; fuel, 100 gals. (oil); 2,000 lbs. coal.



Compressive Power of Road Rollers. — By this is understood the compressive power of the driving roller which sustains the greater portion of the weight. The total weight of this roller and its load is the maximum pressure exerted downward by the machine, and this weight, divided by the width of the tire, gives *the pressure per linear inch of the tire*, which is the effective compression of the surface. For instance a five ton roller with a 39 inch tire will give as much compression as a seven ton roller with a 56 inch tire. Comparison therefore, should be based on the compression obtained and not on the nominal weight of the machine. The compression may be varied by providing driving rollers of different widths. The compression per linear inch of tire, as given for one make of tandem roller is:

Size:	Compression:
2½ ton	100 lbs.
3 ton	140 lbs.
5 ton	210 lbs.
8 ton	265 lbs.
10 ton	275 lbs.

Sizes of Road Rollers.—The usual sizes of tandem rollers are 2½, 3, 5, 8, and 10 tons, and of the three wheel rollers, 8, 10, 12, and 15 tons. This range of sizes has been found to meet the varied requirements of road construction.

FIG. 1,813.—Kelly-Springfield steam steering gear. The two-cylinder engine drives the steering drums through a worm mounted on its crank shaft and meshing with a gear bolted to the drum. By disengaging the small levers controlling the valves of the engine, and placing the worm on the steering column, the machine may be steered by hand. The steam steering gear is used on the 30,000 and 37,000 lb. sizes.

Proportions of Road Rollers.—The following dimensions will serve to illustrate the general practice in the design of machinery of this class:

Case 10 ton three steel roller.

General: Rolling with 81 ins.; wheel base, 113 ins.; length, 18 ft.; height, 10 ft.; ground clearance, 16 ins.; speed, 2¼ miles per hour at normal speed of engine.

Engine: 6⅛ and 9 by 10 ins. compound; 36 brake horse power; Hackworth radial valve gear; fly wheel 36 diameter, 9½ face; revolution, 250 per minute.

Boiler: Locomotive type; diameter, 26 ins.; grate area, 6.14 sq. ft.; heating surface, 130.2 sq. ft.; pressure, 130 lbs.

Erie 5 ton tandem

Weight,
length, 13
width, 4 ft.
el base, 9 ft.
ht to top of
9 ins.

Diameter,
height, 54
ne and one-
ubes.

Two cylinder
re, diameter
ke, 6 ins.

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ns.; width,
ompression,

FIG. 1,814.—Kelly-Springfield rear construction showing two speed sliding gear transmission. The gears are protected by a light steel housing.

CHAPTER 36

FIRE ENGINES

The steam fire engine is practically a portable pumping plant as shown in fig. 1,815; it is in all respects a complete water works on wheels. The boiler, which is generally of the upright semi-water tube type, is combined with the engine by means of a strong iron frame, which carries all the appliances as well as the driver's seat, and also forms the body of the truck. There are usually two vertical pumps of the fly wheel type placed side by side; the exhaust passes up the stack giving the necessary draft.

Some engines are equipped with a boiler feed pump, others depend upon an injector, or feed directly from the main pump. The coal box, which also forms a platform for the engineer to stand upon while underway, is placed back of the boiler. All engines are equipped with two suction and two discharge openings so that connection may be made from either side. The tool box is in front of the engine under the driver's seat. Springs are interposed between the frames and running gear to absorb vibration and shock when running over rough roads or uneven pavements.

Modern fire engines are classified as to size as "double extra first," etc., their capacity and weights are about as follows:

FIG. 1,815.—Continental fire engine. It comprises a vertical semi-water tube boiler, and vertical duplex fly wheel pump. The boiler and pump are attached to substantial horizontal frames which rest on the running gear. A cross head pump and independent pump are provided for boiler feed. The engines are of the slide valve type having yoke connection with the double acting piston pumps.

[illegible]	Capacity	Weight
Double extra first	1,300 gallons per minute	10,800 pounds
Extra first	1,100 " " "	9,800 "
First	900 " " "	8,800 "
Second	700 " " "	7,800 "
Third	600 " " "	6,800 "
Fourth	500 " " "	5,800 "
Fifth	400 " " "	4,800 "

The Boiler.—For fire engine service a boiler is required in which steam may be generated very

FIG. 1,816.—Continental combination shell and water tube boiler. *In construction* there are two cylinders of heavy boiler plate steel having welded seams. Nested together the cylinders are joined top and bottom, to forged spacing rings. The hollow parallel walls are reinforced by stay bolts. The interior tubular grouping or generating system consists of straight boiler tubes made up into sections. These sections or units fill the interior above the fire space and grate. The steam generating system may be described as consisting of two divisions, each being marked by a difference in the size of the tubes and in the manner in which the tubes are disposed. The thin and smaller tubes are inclined to the axis of the boiler, their lower ends converging toward the middle; the heavier and larger tubes are placed vertically and in connection with the two divisions thus formed is a steam separator within the supporting spider at the top. Heat, from the fire on the grate, causes an active upward movement of the water within the small inclined tubes; the current, a mixture, partly water and part steam, impulsively traverses the full length of these tubes. Upon reaching the upper limits of the inclined units, a downward discharge is directed into the top of each large vertical tube. Separation takes place at this point; the steam thus released is taken upward and passes through the top radial headers into the shell.

rapidly, hence water tubes are used; most fire engine boilers are of the semi-water tube type, being a combination of the shell and water tube types. A boiler of this class is shown in fig. 1,816.

It consists of numerous vertical water tubes surrounded by two shells which form an annular space for water and steam. The lower part of the inner shell is enlarged to facilitate the rapid generation of steam by contracting the water space and providing the maximum grate area.

FIGS. 1,817 and 1,818.—End views of Continental combination shell and water tube boiler. Fig. 1,817, top view; fig. 1,818, view as seen from below.

The tubes are divided into outer and inner systems. The outer tubes embrace the short manifold sections adjacent to the fire box walls; they discharge below the water line, and the inner tubes above this point into the steam space. The latter discharge a mixture of steam and water, hence the steam space is by no means dry.

To insure a delivery of dry steam to the cylinders, a peculiar *take off ring* is provided at the highest part of the steam reservoir, the same encircling the inner shell. The ring is perforated with many small holes at its upper part, the steam entering in small streams is held in close contact with the hot inner shell, thus drying it more or less before discharge.

nental boiler running muffler connected and disconnected.

Pumps.—Fire engines are usually fitted with pumps of the vertical double acting fly wheel type. There are two distinct piston pumps united in a single body. The steam and water ends are connected by a frame of steel columns which carry the main bearings for the fly wheel shaft. The steam and water piston of each unit are connected to a yoke, with connecting rod connection with the crank shaft.

Fig. 1,820 shows the usual arrangement of these parts; at the right is a large air chamber and on the other side two vacuum chambers. A small plunger pump for feeding the boiler is attached to the frame and worked from the yoke.

Fig. 1,821 shows construction of the bed plate and frame work, and fig. 1,823 an end view of the pumping engine assembled with valve cover removed for steam cylinder showing the slide valve. The water cylinders are seen with two cylinder heads, and the valve cover removed.

FIG. 1,820.—Continental self-contained pumping unit. *In construction,* the steam and water ends are joined by six steel columns, these being a marine bed plate midway between, by means of which the unit is suspended for the main frame of the steamer. The steam cylinders bed plate, and pumps, each is a single casting.

Hose and Nozzles.—Owing to the contracted diameter of fire hose, the flow of the water is retarded; the loss of power due to friction increases directly with the length of the line and nearly as the square of velocity.

In other words, if the loss due to a given flow be 12 pounds for 100 feet of hose, then 24 pounds will be required to maintain the same rate through an additional 100 feet. To double the velocity will require four times the pressure, or 48 pounds for 100 feet and 96 pounds for 200 feet. It must be evident, then, that the capacity of any engine is diminished as the length of hose is increased.

FIG. 1,824.—Sectional end view of pump cylinder showing location of valves and design of piston. Large removable covers are placed opposite the valves making them easily accessible.

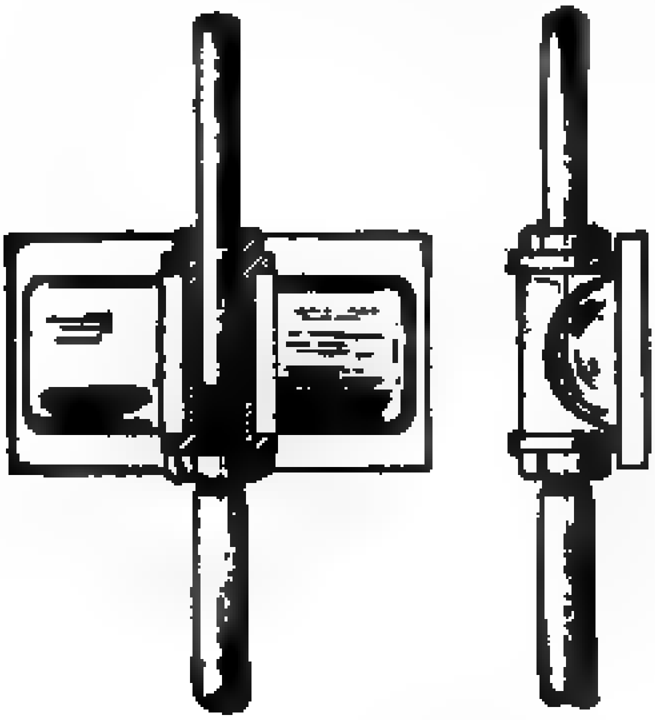
Care should be taken not to use too large nozzles if two or more streams are being thrown. The sizes of nozzles named below will give the most satisfactory results, those in italics being best adapted to fire duty.

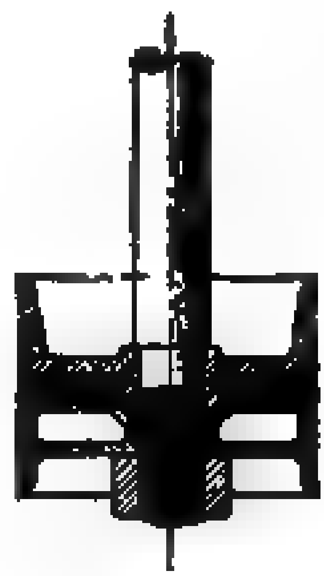
Extra first size engine.—1,100 to 1,500 gallons capacity. Through short lines of hose: *One $1\frac{1}{2}$ inch smooth bore nozzle, for one stream;* one $1\frac{3}{4}$ inch ring nozzle, or one 2 inch ring nozzle; *$1\frac{5}{16}$ inch ring nozzle.* With 1,000 feet of hose, one $1\frac{1}{4}$ inch ring nozzle.

First size engine.—900 to 1,000 gallons capacity. Through short lines of hose: *One 1⅜ inch smooth bore nozzle, for one stream;* one 1½ inch ring nozzle, or one 1⅝ inch ring nozzle; *1¼ inch ring nozzles for two streams.* With 1,000 feet of hose, one 1¼ inch ring nozzle.

Second size engine.—700 to 800 gallons capacity. Through short lines of hose: *One 1¼ inch smooth bore nozzle, for one stream;* one 1⅜ inch ring nozzle, or one 1½ inch ring nozzle; *1⅛ inch ring nozzles for two streams.* With 1,000 feet of hose, one 1⅛ inch ring nozzle.

Third size engine.—600 to 650 gallons capacity. Through short lines of hose: *One 1⅛ inch smooth bore nozzle, for one stream;* one 1¼ inch ring nozzle, or one 1⅜ inch ring nozzle; *1 inch ring nozzles for two streams.* With 1,000 feet of hose, one 1 inch ring nozzle.





FIGS. 1,825 and 1,826.—Plain view and side view of Continental steam slide valve.

FIGS. 1,827 and 1,828.—Pump barrel and piston.

Fourth size engine.—500 to 550 gallons capacity. Through short lines of hose: *One 1¹⁄₁₆ inch smooth bore nozzle, for one stream;* one 1⅛ inch ring nozzle, or one 1¼ inch ring nozzle; *⅞ inch ring nozzles for two streams.* With 1,000 feet of hose, one 1-inch ring nozzle.

Fifth and sixth size engines.—300 to 450 gallons capacity. Through short lines of hose: one 1-inch smooth bore nozzle, for one stream; one inch ring nozzle, or one 1⅛ inch ring nozzle; ⅞ inch ring nozzles for two streams. With 1,000 feet of hose, one ⅞ inch ring nozzle.

Care and Management.—The fire engine is essentially an apparatus adapted to emergencies, and owing to the intermittent nature of the duty performed, it is quite likely, unless the proper precautions be observed, that its several parts, more

Table of Effective Fire Streams

Using 100 feet of 2½-inch ordinary best quality rubber-lined hose between nozzle and hydrant, or pump

Smooth nozzle, size	¾-inch						⅞-inch						1-inch					
Pressure at hydrant, lbs.	32	43	54	65	75	86	34	46	57	69	80	91	37	50	62	75	87	100
Pressure at nozzle, lbs.	30	40	50	60	70	80	30	40	50	60	70	80	30	40	50	60	70	80
Pressure lost in 100 feet, 2½ inch hose, lbs.	2	3	4	5	5	6	4	6	7	9	10	11	7	10	12	15	17	20
Vertical height, feet	48	60	67	72	76	79	49	62	71	77	81	85	51	64	73	79	85	89
Horizontal distance, feet	37	44	50	54	58	62	42	49	55	61	66	70	47	55	61	67	72	76
Gallons discharged per minute	90	104	116	127	137	147	123	142	159	174	188	201	161	186	208	228	246	263

Smooth nozzle, size	1⅛-inch						1¼-inch						1⅜-inch					
Pressure at hydrant, lbs.	42	56	70	84	98	112	49	65	81	97	113	129	58	77	96	116	135	154
Pressure at nozzle, lbs.	30	40	50	60	70	80	30	40	50	60	70	80	30	40	50	60	70	80
Pressure lost in 100 feet, 2½ inch hose, lbs.	12	16	20	24	18	32	9	25	31	37	43	49	28	37	46	56	65	74
Vertical height of stream, feet	52	65	75	83	88	92	53	67	77	85	91	95	55	69	79	87	92	97
Horizontal distance of stream, feet	50	59	66	72	77	81	54	63	70	76	81	85	56	66	73	79	84	88
Gallons discharged per minute	206	238	266	291	314	336	256	296	331	363	392	419	315	363	406	445	480	514

especially its interior mechanism, will suffer more deterioration while standing idle than from actual service.

The following instructions as given by the American-La France Fire Engine Co. will be found very helpful in acquiring a proper understanding of the fire engine and its management.

General Directions.—All things about the house should be kept in good order and neat condition, particularly the engine, which ought always to be clean and bright. Avoid neglect, which tends to waste and decay, as dirt often covers unsuspected faults.

While standing in the house, the engine should at all times be kept ready for immediate service, with plenty of shavings and kindlings in the fire box, and as much kindlings and coal in the fuel pan as can be conveniently carried.

In winter, if no heater be attached to the engine, the room must be kept warm, to insure against frost.

FIG. 1,829.—Siamese connection. This is used for stand pipes attached to the outside of buildings, etc., and also as an appurtenance of the fire engine, as for leading off two lines of hose.

FIG. 1,830.—Approved form of strainer. This should be attached to the bottom of the suction pipe.

The joints and connections in the suction must be perfectly tight. The stuffing boxes of the engine and pump should be well packed.

All of the bearings and journals, as well as the oil cans, should be well supplied with good oil. The best lard oil is recommended for this purpose, and in winter it should be mixed with kerosene, in the proportion of two parts of the former to one of the latter, to prevent its becoming thick.

From three-fourths to one inch of water should be indicated in the glass gauge, except there be a heater attached to the engine, when from four to five inches should be carried. The bottom of the glass tube being on a line with the crown sheet, when one inch of water shows in the tube the water line in the boiler is then one inch above the crown sheet.

Every engine house should be provided with a force pump, for filling the boiler with water, as well as for washing and other purposes, fitted for one inch hose, together with at least 25 feet of the latter.

It is advisable occasionally, say once a month, in towns where fires are not frequent, to take the engine out for practice and drill, and to make sure

that it is in proper working order, after which the boiler should be blown off and refilled with fresh water, as hereinafter directed.

Laying of Fire.—Before laying the fire, see that the grate and fire box are clean, also that the grate bars are fast, so they will not be liable to jar out, and that all the steam outlets of the boiler are tightly closed.

Lay on the grate some dry pine shavings, not too many, spread evenly over the grate, with a few hanging down between the bars; on the shavings put some finely split pine or hemlock wood, then some a little coarser, and finally a quantity coarser still. It is well to put on the top some finely split hard wood. These kindlings must all be dry and split, not sawed, and should be put in loosely, in layers, the layers being crossed, so that there will be a free circulation of air between them.

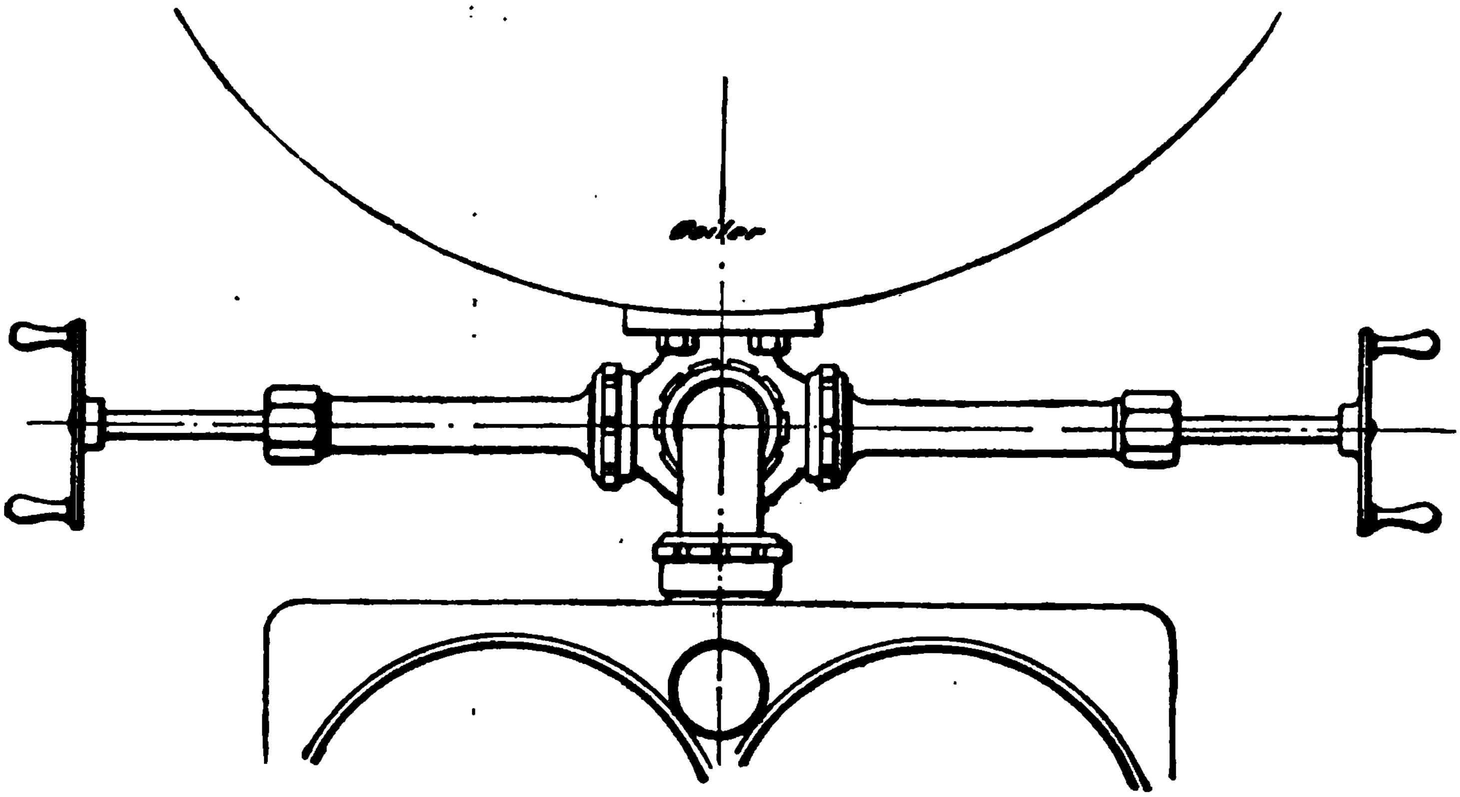


FIG. 1,831.—Continental duplex throttle. ***Its object*** is to allow the operator more freedom and to permit a wide radius of action about the machine. ***It consists of*** two complete and distinct valves joined in a single body. The operating stems are extended to opposite sides of the engine within easy reach. Both stems turn to open or close in the customary manner for respectively starting or stopping.

Never start a fire unless a full gauge cock of water is in the boiler.

Always keep a good torch in the fuel pan, ready for use. This can be made by taking a stick about two feet long and tying some cotton waste on one end, and saturating the waste with kerosene.

Starting and Maintaining the Fire.—To light the fire: Apply torch, already described, *below the grate*, never in the door; and while doing so move the torch around to insure thoroughly igniting the shavings.

Be particular not to open the fire door oftener than necessary, especially when getting up steam.

In addition to the wood in furnace, an extra supply is carried in the fuel pan. For convenience this is put up in bundles; their size should not

exceed four or five inches in diameter, nor much over a foot in length. If these dimensions be exceeded they become cumbersome and cannot be readily passed into the furnace.

The kindling should be carefully prepared, and the quantity carried sufficient to generate a working pressure in the boiler before coal is added to the fire.

When there is a pressure of 40 to 60 pounds of steam, begin throwing in coal, a little at a time, broken up in pieces about the size of a man's fist. Bituminous coal should be used, the same as that from which illuminating gas is made. It should be of the very best quality, entirely free from slate or sulphur.

FIG. 1,832.—Continental steam blast valve control. By opening this valve an artificial draft may be started in the chimney for trimming the fire at such times when the engine mechanism is at rest.

The mistake is frequently made of allowing most of the wood to burn out before putting in any coal. This should be avoided, as the kindlings must be burning nicely in order to start the coal. If the supply of wood should become exhausted, always begin throwing in coal while there is still enough wood in the fire box to ignite it, even if the gauge do not indicate any steam whatever.

Do not put the wood or coal all close to the fire door, but scatter it about and spread it evenly over the grate.

As soon as the engine is started, coal should be put in often, a little at a time, and the grate should be kept nicely covered, but not thickly—say to a depth of three or four inches. Be particular to fire evenly and regularly, taking care that there be no air holes through the fire, and to open the fire door only when necessary.

The grate bars should be kept well raked out from below, and the fire and coal occasionally stirred off the grate bars inside the fire box, using the flat side of the poker for the latter operation. But do not "clean" oftener than necessary; keep the fire door open as short a time as possible, and use no more coal than is required.

Operating the Engine.—The engineer should start up the machine gradually, but before doing so he ought to satisfy himself that the joints and connections in the suction hose are air tight, that the discharge gate is open and the churn valve closed, and that the fire has been properly attended to. Let the cylinder cocks be open and the exhaust nearly closed, and all the bearings and journals well oiled. The wheels should be properly blocked, especially if standing on a grade. When starting, the throttle valve should be opened slowly at first, or condensed steam will be thrown out of the stack on the dome, and is liable to stain it.

The automatic air cocks on the upper pump heads must be opened immediately after starting. They serve to promptly relieve the upper pump chambers of air, and may be closed as soon as water is ejected from their orifices.

When condensation has ceased, the engine being warm, the drain cocks should be closed and the machine speeded up gradually until a good pressure of steam is obtained.

After the engine is fairly started, do not stand too close, but let your position be a step back; and, with your face toward the machine, endeavor to train your eyes and hands to command the entire situation. While it is perfectly proper to be near the throttle, in order to promptly close it in case of bursting hose or failure of the water supply, do not acquire the habit of constantly clinging to the same, for there are other duties equally as important that require your attention.

In the general hurry and rush, avoid all excitement, and let your duties be attended to in a calm and collected manner.

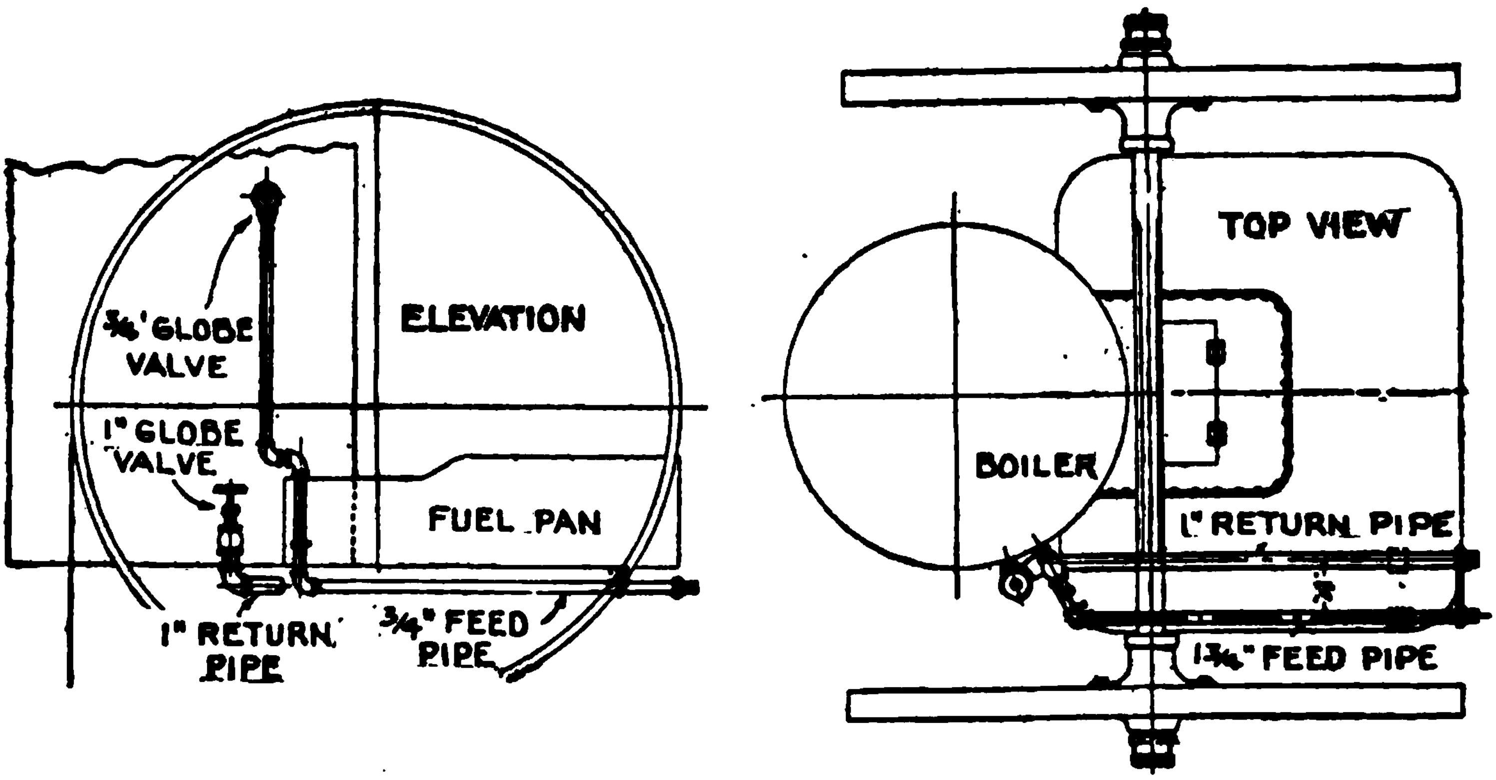
Until the engineer has had some experience with the machine, and is thoroughly familiar with its workings, it is not advisable for him to use more than 90 or 100 pounds of steam, which is all that is required for ordinary fire duty, and the necessity for more than 120 pounds will never arise.

The water in the boiler should be carried as high as six or eight inches in the glass tube as soon as the engine is fairly at work and a good pressure

of steam is obtained. The gauges will indicate more water in the boiler when the machine is running than it will with the same quantity of water if it is not at work. For this reason, the boiler should be kept well supplied with water; and the feed tank (when there is one) should be kept always full.

If the glass tube get broken, the height of the water line in the boiler can be ascertained by means of the gauge cocks, opening them but a trifle. If opened wide the tendency would be to draw up the water. A little experimenting will enable one to use them properly, but sole reliance should not be placed on the glass tube at any time, and the gauge cocks ought to be used frequently. Both the gauge cocks and the glass water gauge should be kept clean, and water from the glass gauge should be blown out occasionally.

The uniformity of the boiler's action is materially aided by maintaining an even fire and a steady feed.



FIGS. 1,833 and 1,834.—Usual arrangement of heater pipes on Continental engine. The pipes are fastened to the bottom of the fuel pan and the spacing and dimensions as given conform with the arrangement of pipes for the stationary heater shown in fig. 1,835.

If there be a tendency to foam, the feed should be increased and the surface blower opened quite frequently to relieve the boiler of the scum and surplus water. If the foaming be unusually violent, it may be subdued by stopping the engine for a few moments and permitting the water to settle.

During temporary stops the fire should be cleaned, the clinkers removed, and the moving parts of the machinery examined and oiled.

When the engine is not running, the fire may be replenished if necessary; and it can be kept bright by slightly opening the blower valve, by means of which a steam jet is blown into the smoke stack for the purpose of improving the draught.

Boiler Feed.—The boiler is usually fed by force pumps, the plungers of which are secured directly to the yokes of the main engines. Both pumps are arranged to work in unison; and the supply is generally taken from the discharging chamber of the main pumps, and is controlled by an ordinary globe valve.

Should the water being delivered by the main pumps be unsuitable for feeding the boiler, this valve must remain closed, and a supply from some other source introduced through the opening provided for that purpose.

Every engine required to pump salt water, or other water unfit for the boiler supply, should be provided with a fresh water feed tank.

The purpose of the automatic air cock (if there be one) is to prevent the rattling of the check valves when the pumps are being only partially filled; if the supply is to be draughted from a barrel or tank, the entrance of air through this cock must be prevented.

In any engine, to feed the boiler directly from the main pump, the water gauge must indicate a greater pressure than the steam gauges; and it may be necessary, in order to obtain the desired water pressure, to partially close the discharge gates of the pump if a large nozzle or two or more streams are being used.

When feeding the boiler, it is a good plan to occasionally feel the pipe leading from check to boiler with the hand, as one can tell by this means whether the pump be feeding properly. If feeding all right, the pipe will be cool. If the pipe be hot, the pump is not feeding properly, and it should be attended to.

In case of low water, and it is found impossible to feed the boiler in any of the different ways provided, the fire must be drawn at once. Don't turn on the feed, start or stop the engine, nor open the safety valve; let the steam outlets remain as they are, and allow the boiler to cool down.

Relief Valve.—This is a device attached to the discharge main of the pump, and connecting with the suction chamber, its purpose being to relieve the hose of undue pressure. It is used in connection with a shut off nozzle. When such nozzle is either partially or fully closed the valve is operated automatically, like a safety valve, and the surplus water not required for the duty being performed is diverted from the hose into the suction chamber of the pump, without any cessation to the machinery. Its operation is similar to that of a churn valve, the difference being that the relief valve works automatically while the churn valve does not.

If the engine be supplied with a relief valve you should familiarize yourself with its construction and working. It is not practicable for us to give directions here that will apply to all the several different types of relief valves now in use, most of which are no longer made. Care should be taken, however, whatever the style of valve, to see that *all connections*

Protection Association, etc. The 2½ inch hose coupling is the size most generally used by public fire departments. The 3 and 3½ inch sizes are used mainly for high pressure or fire boat services and are not in general use. For sizes under 2½ inches, there is no universal standard; there are at least six different so called "standards" used, known as follows: Eastern gauge hose thread (used in the New England States), Pacific Coast hose thread, known also as the California standard hose thread (used on the Pacific Coast); Chicago hose thread (used in the Middle West); Pittsburg hose thread; Boston hose thread, and the iron pipe thread, which is the general standard for pipe threads. In addition to these "standards," there is a great diversity of 2½ inch threads used by the fire departments of various cities. As regards the standards, there is no absolute agreement as to the dimensions. The only actual standard is that of the iron pipe thread, which, of course, is extensively used in all parts of the country.

The following threads have been adopted by National Board of Fire Underwriters, American Water Works Association, New England Water Works Association, National Fireman's Association, National Fire Protection Association.

Size	2½ inch	3 inch	3½ inch	4½
Threads	7½	6	6	4

are kept tight, in order to prevent any leakage of air into the suction chamber of the pump.

There ought always to be a valve between the relief valve and the suction chamber of pump, so as to cut out the relief valve in case same should from any cause become disabled.

Priming Valves.—The priming valves, in cases where such valves are attached to a fire engine, control small passages leading from the discharging side of the main pumps to the upper receiving chambers of the same. If the air cocks fail to show water promptly, flood the upper pump ends by opening these valves for a moment, provided of course, that the lower ends of the same have already taken suction.

The Variable Exhaust.—In connection with good coal and good firing on the part of the stoker, the engineer must make proper use of the exhaust lever, to maintain an ample working pressure of steam. When there is plenty of steam the exhaust should be kept wide open; if more steam be required push in the lever. This will diminish the opening, and the velocity of the exhaust will be increased, improving the draught, but creating a back pressure on the engine.

The variable exhaust is particularly useful when first starting, but as the boiler steams more freely open it to the full extent.

The Churn Valve.—The principal object of the churn valve is to permit the operation of the pumps without discharging any water through the natural channels; it controls a passage by which the discharging side of the pumps is connected with the suction chamber.

In draughting water, when the pumps are first started, *this valve must remain closed* until the pumps are filled with water, thereby excluding the air which would find its way into the suction chamber if the same were open. It should also be closed when the pumps are at rest, to prevent the dropping of the water in the suction pipe. It may be opened slightly with good effect when pumping through long lines of hose, or when first starting against a heavy resistance, thereby increasing the piston speed of the machinery without actually delivering a greater quantity of water. It also permits the force pumps to be kept in motion, for the purpose of supplying the boiler at times when it is undesirable to deliver water through the hose lines.

When the engine is put to suction, acquire the habit of feeling this valve to assure its complete closure.

The Water Supply.—The suction basket or strainer should always be attached when draughting water, and every precaution taken *to insure tight connections in the suction.* The basket must be kept well under the

surface, and kept from clogging if the water be foul. Additional strainers should be provided, placed just inside of the suction inlets of the pump when the suction is carried disconnected; when the suction is permanently connected to the pump, the strainer is set in the end of the suction. These strainers must always be examined and cleaned before the engine is returned to quarters, and at all other times when there is any reason to suspect that they are obstructed.

When the supply is taken from a hydrant, satisfy yourself that the same has been fully turned on; if opened before water is wanted through the hose, the discharge gates on the engine must be closed. Unless the pressure be excessive, the hydrant is usually permitted to remain open while the

FIG. 1,836.—Front view of Continental fire engine showing method of withdrawing wire strainer for inspection. The strainer intercepts foreign particles which would otherwise clog the valves of the pump. The strainer may be withdrawn from either side as shown without lifting, disconnecting or otherwise disturbing the suction hose.

steamer is attached, the discharge during temporary stops being controlled by the engine gates.

Frequently the first water taken from a hydrant is stagnated; hence, if necessary to feed the boiler before any considerable quantity has passed, it is advisable to permit it to waste by opening an idle gate.

The apparatus should always be halted, or placed at a proper point, with reference to the source of the water supply. Good judgment on the part of the driver will often obviate short and awkward bends in the suction hose, and also facilitate the work of making the necessary connections. The suction hose should always receive considerable attention; oil is very injurious to the rubber, and when allowed to remain long in contact with its surfaces, will cause decay. Ordinary precaution will be sufficient to prevent injury by chafing on sharp stones or rough surfaces when the pumps are in operation.

When attached to a hydrant or plug, do not run the engine faster than you can get water to supply the pump, and if the pressure be not sufficient to allow the pump to work to its full capacity, avoid using too large nozzles.

If it be suspected that one of the joints in the suction is loose, the speed of the engine may be slackened without stopping entirely, until water is thrown eight or ten feet from the nozzle, when if the pump be taking air, the stream will crack and snap instead of flowing out smoothly. If it be found that the pump is taking air through the suction, and the leak cannot be located in any other way, it may be found by removing the suction basket and turning the end of the suction up higher than the top of the pump, and then filling it with water. The water will be forced out through the joints wherever loose, and leaks can easily be found in this way.

When draughting the water, bear in mind that the greater the perpendicular lift, the less the quantity of water which can be pumped, remembering that it is the pressure of the atmosphere which forces the water into the pump, and not any power exerted by the pump itself, which simply produces the vacuum. Thus, the nearer the surface of the water the greater the velocity with which it enters the pump, while the higher the pump, the weaker the pressure and the less the quantity of water which enters it, and at a height of about 30 feet no water will go in the pump.

For this reason, the greater the lift, the smaller the stream that can be thrown effectively, and the size of nozzle used should depend upon the height the water is draughted, reducing it one-eighth inch for every five feet above a lift of 10 feet.

Shutting Down.—In extremely cold weather, if it be desired to stop doing duty for any reason, it is a good plan to keep the main pump constantly but slowly revolving, even if it be just barely moving, and to keep a light feed on the force pumps, to prevent freezing. This should be done, also, when necessary to change positions at fires, while the engine is being transferred. In small towns it is well to have a good supply of water in the boiler, and sufficient steam to revolve the engine and pump slowly while returning to the house.

Preparatory to the final shutting down of the apparatus, with a view of returning to quarters, permit the steam pressure to rise to the point of blowing off; also let the fire be burned *clear and bright before withdrawing*

the same from the furnace, which may be readily done by closing the fire door and opening the blower valve. This will burn off most of the soot adhering to the heating surfaces. Allow water to drop down to first gauge cock, which will insure your obtaining dry steam; when blower is opened with a high water line, water is apt to rush through the blower, and wet steam is not so effective in blowing off the soot.

There should be a steam pressure of about 30 pounds when the grate is dumped, after which all remaining soot and ashes should be blown with *steam* from the top of the smoke stack down through the smoke flues into

FIG. 1,837.—Front end of Ahrens-Fox, self-propelled fire engine with gas engine power unit; view showing the power and pumping unit. The gas engine is used both for propelling the machine and for pumping.

the fire box, using for this purpose the small cleaning hose. Then the soot should be blown from the fire box and water tubes, and the ashes from around them, using the cleaning hose and steam through the fire door; the grate may then be replaced.

Not less than once a month, after dumping the grate, with from 15 to 20 pounds of steam, all of the water and steam should be blown out of the boiler through the blow off cock, in the water leg of the boiler.

Returning to Quarters.—Promptly refill the lubricators and all other oil cups, and thoroughly examine the mechanism, and also the running gear, as soon as the apparatus is returned to its quarters. If, however, the work has been of long duration or the water bad, the boiler should be thoroughly washed out before again placing the machine in service. To do this properly, a one-inch hose with suitable nozzle must be provided, and if there be no hydrant connection or pressure, a force pump may be substituted. Remove all the plugs at the bottom of the boiler, and with the hose and scrapers free the shell of the sediment lodged therein.

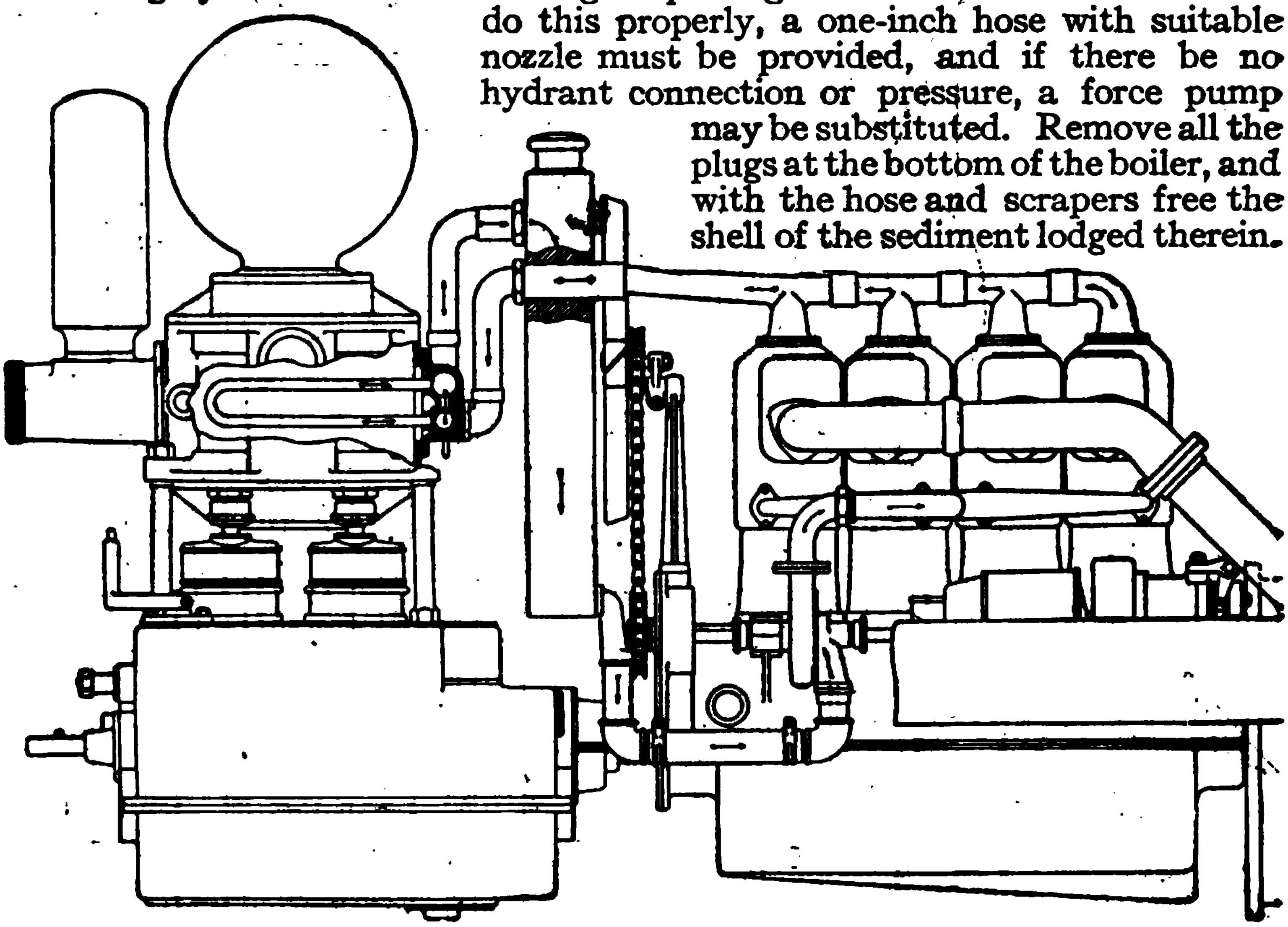
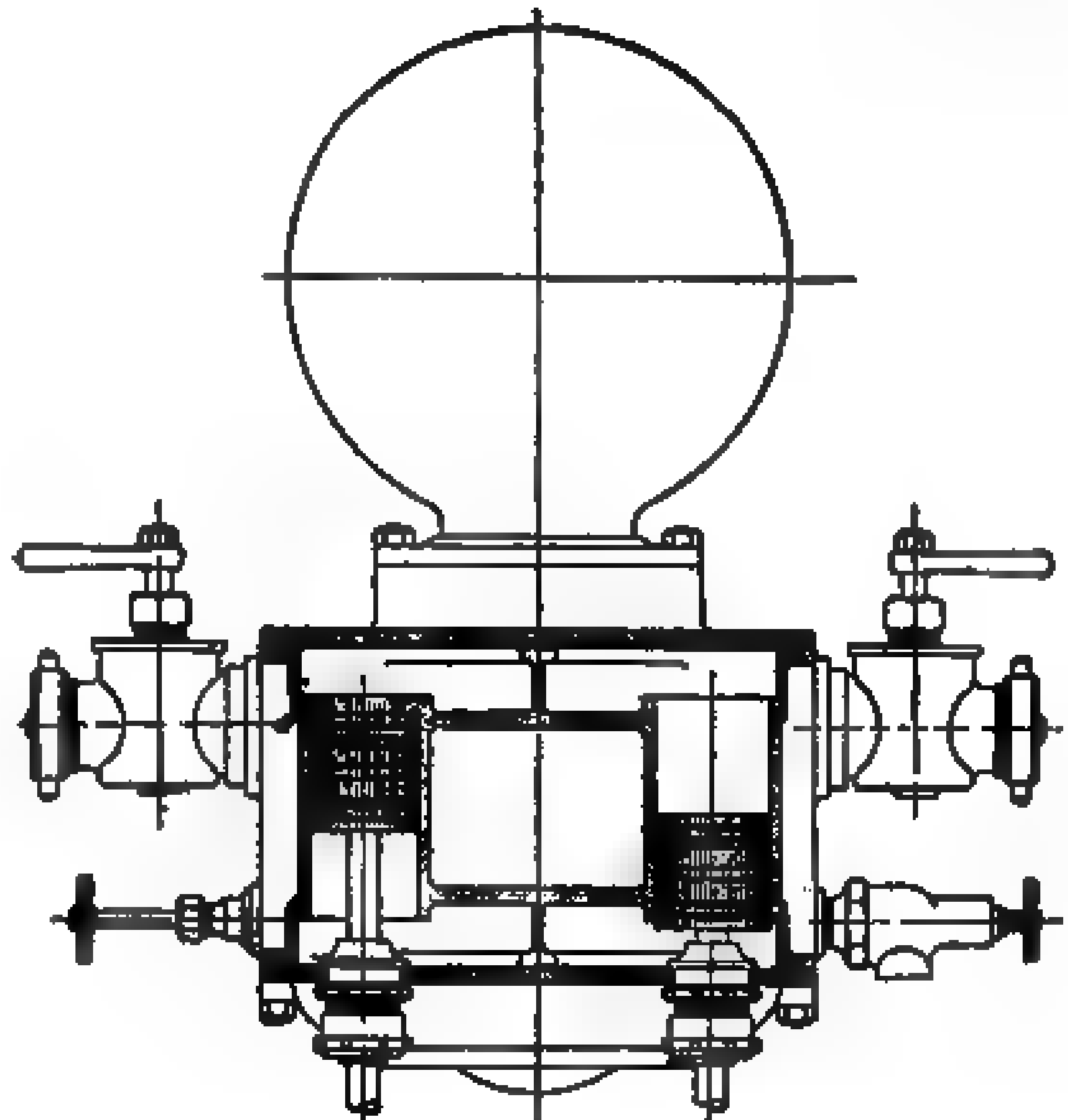


FIG. 1,838.—Sectional view of Ahrens-Fox six cylinder engine and pumping unit, showing cooling system. ***The cooling system*** includes a junction box seen at the rear of the pump. To the upper and lower divisions of this junction box are affixed a number of tubular loops made of thin copper. It is seen that these several loops penetrate well into the depth of the suction chamber. Their exterior surfaces, therefore, are subjected to the cooling influence of the full volume of water discharged by the fire pump, all of which must necessarily pass through the chamber where the loops are installed. A two way regulating sleeve, located in the lower division of the junction box, affords control of the device. Turned to the position appearing on the diagram the sleeve has cut off entrance to the cooling loops. The hot flow leaving the top of the motor cylinders will therefore by pass into the radiator, thus following the circulative course ordinarily prescribed. Reversing the position of the sleeve the flow from the motor is diverted and forced to pass through the loops. In transit the exterior surfaces, in contact with the rapid rush of water, absorb heat very rapidly and after making the loop circuit the flow proceeds to the radiator and resumes the course which ordinarily obtains. The flow can be diverted to the loops in any degree desired by the operator. When driving over the road the hot flow from the motor is prevented entering the tubular loops by setting the lever in its off position. Unnecessary radiation of heat in the interior of the pump is thereby avoided. The cooler may also be used as a heater, that is by keeping the loops in circuit at times when extremely cold weather is encountered, the pumps may be kept quite warm as long as the motor is in action. This feature is noteworthy and is of advantage in cold climates. Enroute, coming or going, the pump will be protected from the danger of freezing.

FIGS. 1,839 and 1,840.—Elevation and plan of Ahrens-Fox, self-propelled fire engine with four cylinder gas engine drive, showing general arrangement of the mechanism.



FIGS. 1,841 and 1,842.—Sectional view of Ahrens-Fox twin double (four cylinder) piston pump. Fig. 1,841 cross section through center; fig. 1,842 cross section through water cylinders.

While the water is out of the boiler, examine the stop cocks on the ends of the glass water gauge, and see that their openings are clear; always coat their surfaces with cylinder oil when replacing, and adjust them to be easily closed should the glass be broken.

After pumping dirty or salt water, the pumps should be emptied and well rinsed, and then refilled or primed.

After every working, and while the parts are still warm, pour a small cup of good cylinder oil into each

FIGS. 1,843 and 1,844.—Sectional views of Ahrens-Fox twin double (four cylinder) piston pump driving mechanism. Fig. 1,843, transverse cross section; fig. 1,844, longitudinal cross section showing crank, clutch, etc.

of the oil cups on the top heads of the engines; have the pistons at a point preventing the oil entering the ports, and after allowing sufficient time for the same to distribute itself over the piston head, give the engines several turns by hand, thereby coating the sides of the cylinders with a film of oil and effectually preserving them against rust.

It is a matter of the greatest importance that all of the joints and connections in the suction should be kept tight at all times. Every little while the engineer ought to take the wrench furnished for that purpose, and see that every joint in the suction is provided with a piece of good packing and that it is perfectly tight.

The steam gauges should stand at zero when pressure is off.

If necessary to clean the glass tube in the water gauge, close the cocks on top and bottom, fill the tube with benzine, and allow it to stand an hour or two. Then draw the benzine out, open the cocks, and let water in again.

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